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PROPELLERS WITH ADJUSTABLE BLADES.

Results of Model Experiments.

By Dr. H. F. Nordström*.

Introduction.

The purpose of this paper is to give an account of some tests with propellers with adjustable blades which have been carried out during 1943 and 1944 at the Swedish State Shipbuilding Experimental Tank at Göteborg.

The tests were ordered by Consul General Axel Ax:son Johnson. The technical programme for the research work has been prepared by a committee consisting of Mr. Elov Englesson (designer of the Kamewa Propeller), Mr. Folke Seldén (Consulting Naval Architect), and the author of this paper.

When designing a propeller with adjustable blades, working under the most varying load conditions such as towing and ice breaking, the question arises as to whether the initial pitch (the

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pitch for which the propeller is to be designed) should be determined according to conditions when free of tow or be made suitable for conditions at additional load, and thus be made with reduced initial pitch. It is assumed that the same propeller diameter is used in all cases. With reduced initial pitch it can be foreseen that the performance when working astern will be improved if reversing is assumed to take place by adjusting the blades without altering the direction of rotation. It is, in the first place, these questions which will here be investigated. It does not appear that any investigation of questions of this kind has been made before. The matter has now, however, a certain current interest since propellers with adjustable blades have come into an increasing practical use. Such an investigation may also be of importance in connection with the use of propellers with adjustable blades for ships in general.

In connection with these investigations the question of the influence of the propeller Rpm. has been studied. A comparison has also been made between propellers with adjustable blades and propellers with fixed blades.

Symbols and Units.

The following symbols, adopted by the Conference of Tank Superintendents in Paris, 1935, are used.

∇ = displacement (volume)
N = shaft horsepower (SHP)
n = revolutions per unit time (Rpm. or Rps.)
D = diameter of propeller
H_0 = initial face pitch
H = pitch in general or adjusted pitch, when adjustable blades are used
H_0/D = H/D = } pitch ratio
Q = torque
$C_Q = \dfrac{Q}{\varrho D^5 n^2}$ = torque coefficient
T = thrust
$C_T = \dfrac{T}{\varrho D^4 n^2}$ = thrust coefficient
ϱ = density of water
η = propeller efficiency (in open water)

P = pull
V = speed (speed of ship or speed of advance)
$\lambda = \frac{V}{Dn}$ = advance coefficient
g = acceleration due to gravity

Metrical units are used throughout.

1 ton = 1 000 kilos (1 ton = 0.984 Engl. tons)
1 knot = 1 852 m/h (1 knot = 0.999 Engl. knots)
1 HP = 75 mkg/s (1 HP = 0.986 Engl. HP)

Tested Ship Model.

As the object to be experimented on, a model of a harbor ice-breaker, designed by Mr. Folke Seldén, was used. The model, which is hereafter referred to as No. 82, was made of wood and to the scale 1 : 12. The oultines can be seen from Fig. 1. Fig. 2 is a photograph of the model during the erection of a propeller.

The main data of the ship and model were as follows:

	Ship	Model
Length on WL.....	= 31.7 m	2.64 m
Length between PP....	= 32.5 m	2.71 m
Breadth on WL.....	= 9.4 m	0.78 m
Mean draught	= 4.0 m	0.33 m
Displacement	= 535 m³	309.6 kilos (in tank water)

Resistance tests and self propulsion tests were made in smooth water and with the same immersed model volume.

Tested Propeller Models.

Altogether six sets of blades were tested, there being three blades in each set. The same boss was used for all sets. The propeller blades as well as the boss were designed in collaboration with Mr. Hans Edstrand (at the Karlstads Mekaniska Werkstad, Kristinehamn).

With five of these sets (P49 to P52, and P55) the available SHP (at the shaft end) was, in the first place, assumed to be 1200 at 154 Rpm. The initial pitch for P49 was calculated to suit conditions when free of tow. The initial pitch for P50 and P51 amounted to 2/3 and 1/3, respectively, of the initial pitch for P49. P52 had initial pitch = 0, *i. e.* the blades were plane on the

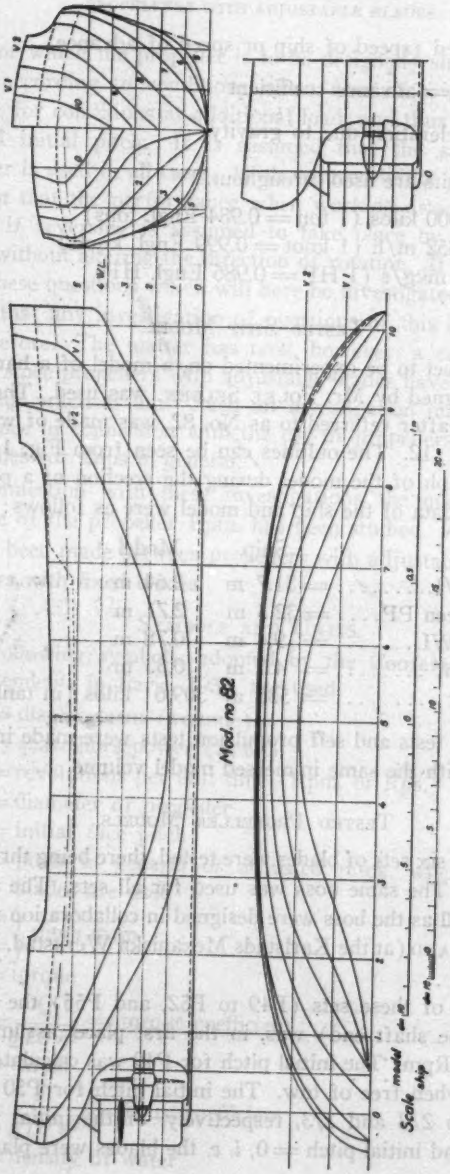
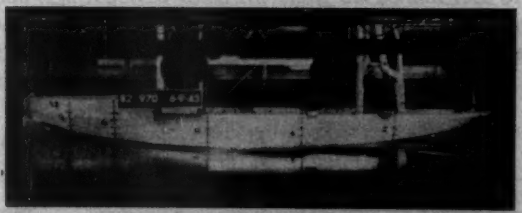


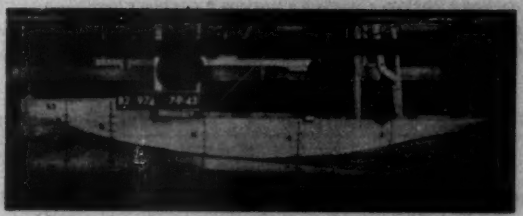
Fig. 1

Fig. 1a

FIG. 2.



10.3 KNOTS.



12.7 KNOTS.

FIG. 5.

driving face (when going ahead). P49—P52 had a constant initial pitch on the driving face. P55 was identical with P49 except that the initial pitch was somewhat reduced towards the boss. All the propellers had the same diameter, same expanded area, and the same sections. The diameter will, of course, be different according to whether the calculations are based on conditions corresponding to free of tow or heavy towing (see appendices 1 and 2). In the latter case a larger diameter is required. In the present case, as often happens, the diameter was determined with consideration to the space in the propeller aperture. The diameter chosen is, on the whole, the most suitable one for free of tow. Fig. 3 is a drawing showing the propellers.

The sixth propeller, P82, was calculated for a propeller speed (231 Rpm.) which exceeded by 50% that of the previous propellers. The same SHP, *i. e.* 1200, was assumed. As the same boss as for the other propellers was to be used the diameter was chosen somewhat larger than would otherwise have been the case. The initial pitch, which was kept constant, was chosen about 10% smaller than that required when travelling ahead and free of tow. The shape of the blades can be seen from Fig. 3. The sections were of the same character as for those of the other propellers.

The principal data for the six propellers can be seen from Table 1

Table 1. Particulars of Tested Propellers.

Propeller	P49		P50		P51		P52		P55		P82	
	m	*s*	*m*	*s*	*m*	*s*	*m*	*s*	*m*	*s*	*m*	*s*
Diameter D ..... mm	300	3 600	300	3 600	300	3 600	300	3 600	300	3 600	260	3 120
Initial pitch H_0 mm	207	2 484	138	1 656	69	828	0	0	207	2 484	120	1 440
H_0/D	0.69		0.46		0.23		0		0.69		0.46	
	1		$^2/_3$		$^1/_3$		0		1		—	
Number of blades	3		3		3		3		3		3	
Exp. area/discarea %	40		40		40		40		40		35	

m = model, *s* = ship

At another setting than the initial pitch, the pitch was characterized by the distance in the axial direction between two points on the blade at a radius = 2/3 of the outer radius and at an angle of opening viewed in the axial direction = 40°. The expression "adjusted pitch" or simply "pitch" will be used in this con-

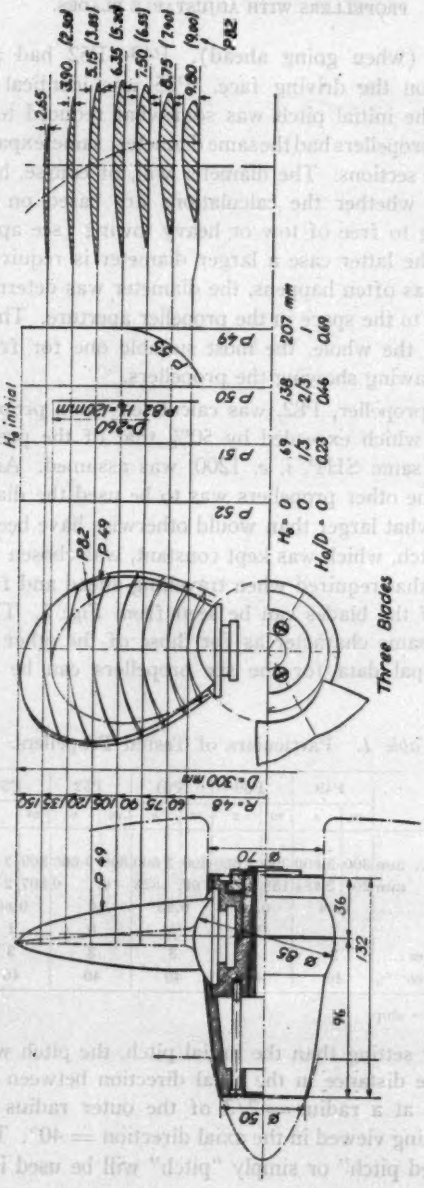

(2 m)
(3 m)
(5 m)
(6 m)
(7 m)
(9 m)
P 82
2.65
3.90
5.15
6.35
7.50
8.75
9.80
P 49
P 50
P 51
P 52
207 mm
1
0.69
138
2.5
0.46
69
1.5
0.23
0
0
H_0 initial
H_0/D = 0
D = 260 mm
P 82 H_0 = 130 mm
D = 300 mm
R = 45 60 75 90 105 120 135 150
70
36
32
85
96
50
Three Blades

Fig. 3

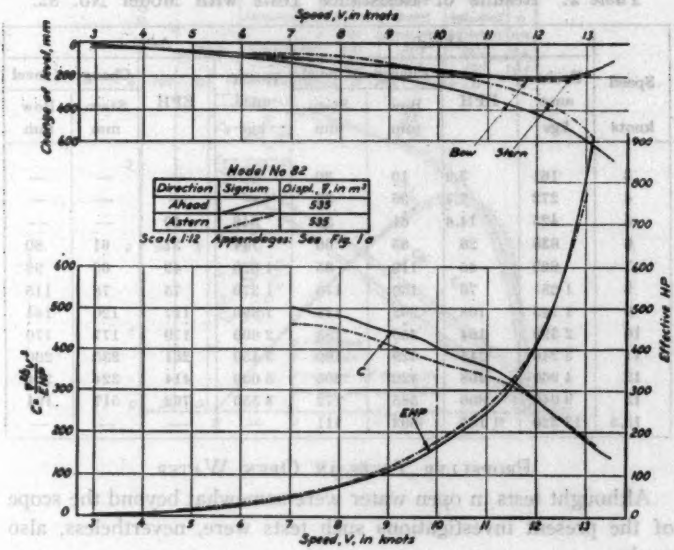


Fig. 4

nection, although the word "pitch" in the strict sense is not justified.

Each individual blade could be adjusted by hand and then locked (see Fig. 3).

Resistance Tests.

With the model resistance tests were made within a range of speed corresponding to 3 to 13.5 knots. In these tests the model was provided with a rudder and a blind boss according to Fig. 1a. The results of these tests can be seen from Fig 4, also from Table 2. The primary results with the model were transferred to the scale of the actual ship in the conventional way according to Froude's method. The weight of the water was assumed to be 1.015 tons/m³.

In Fig. 5 photos of the model when travelling ahead at two different speeds are presented.

Resistance tests were also made with the model when going astern. The model was then in the same condition as during the tests when going ahead. The results of these tests can also be seen from Fig. 4 and Table 2.

Table 2. **Results of Resistance Tests with Model No. 82.**

Speed	Ahead				Astern			
	Resistance	EPH	Change of level		Resistance	EPH	Change of level	
			Bow	Stern			Stern	Bow
knots	kgs		mm	mm	kgs		mm	mm
3	161	3.3	10	30	—	—	—	—
4	272	7.4	36	36	—	—	—	—
5	425	14.6	61	50	548	19	—	—
6	638	26	85	66	744	31	61	80
7	929	45	119	85	1 020	49	66	98
8	1 281	70	160	115	1 370	75	78	115
9	1 745	108	202	138	1 890	117	120	144
10	2 390	164	256	158	2 600	179	177	170
11	3 240	245	329	190	3 450	261	235	200
12	4 960	408	420	205	5 030	414	324	224
13	9 040	806	585	172	8 550	762	517	194
13.5	12 320	1 141	693	111	—	—	—	—

Propeller Tests in Open Water.

Althought tests in open water were somewhat beyond the scope of the present investigations such tests were, nevertheless, also made.

Fig. 6 shows the results of tests in open water with P49 and P50, whereby the pitch in both cases was adjusted to the initial pitch of P49. In Fig. 7 the results of tests with P49, P50, P51, and P52 are shown with the pitch in all cases adjusted to the initial pitch of P50, and in Fig. 8 the results with P49 and P51 are given with a pitch equal to the initial pitch of P51. Finally, in Fig. 9 the results with P49 and P52 are shown with the pitch adjusted to 0, *i. e.* the initial pitch of P52.

Self Propulsion Tests.

The main part of the research consisted of tests with the model self propelled by the different propellers, partly when travelling free of tow ahead and astern, partly when towing ahead at 7, 3.5, and 0 knots and astern at 0 knots.

Altogether, with the resistance tests and the propeller tests in open water, 146 series of tests with the different propellers were made at different settings of the pitch. Altogether 725 runs were made with the towing carriage. Very extensive tests results are in hand which have not yet been completely worked out. Of the results obtained so far, only a selection can be presented here.

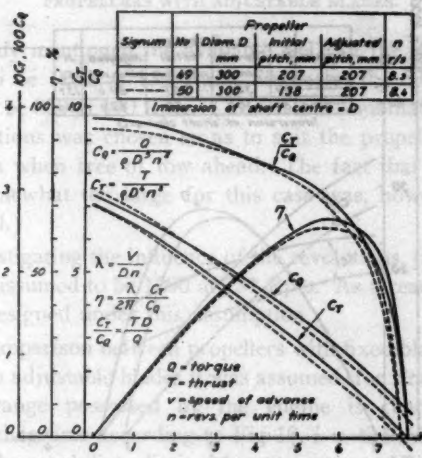

Propeller
Signum | No | Diam. D mm | Initial pitch, mm | Adjusted pitch, mm | n r/s
49 | 300 | 207 | 207 | 8.3
50 | 300 | 138 | 207 | 8.4
Immersion of shaft centre = D
10Cq, 100Cq
η, %
CT/Cq
Cq = Q / ρD⁵n²
CT = T / ρD⁴n²
λ = V / Dn
η = λ/2π · CT/Cq
CT/Cq = TD/Q
Q = torque
T = thrust
v = speed of advance
n = revs. per unit time
10λ

Fig. 6

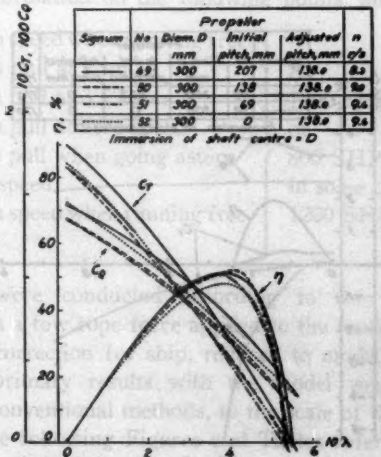

Propeller
Signum | No | Diam. D mm | Initial pitch, mm | Adjusted pitch, mm | n r/s
49 | 300 | 207 | 138.0 | 8.3
50 | 300 | 138 | 138.0 | 8.4
51 | 300 | 69 | 138.0 | 9.6
52 | 300 | 0 | 138.0 | 9.6
Immersion of shaft centre = D
10CT, 100Cq
η, %
CT
Cq
η
10λ

Fig. 7

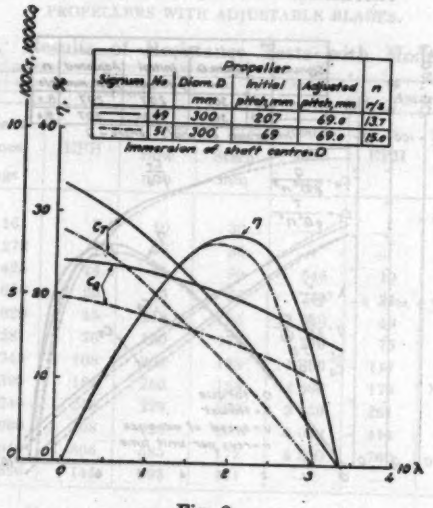

Propeller
Signum | No | Diam. D mm | Initial pitch, mm | Adjusted pitch, mm | n r/s
P49 | 300 | 207 | 69.0 | 13.7
P51 | 300 | 69 | 69.0 | 15.0
Immersion of shaft centre = D
100 C_T, 1000 C_Q
η %
C_T
C_Q
η
10λ

Fig. 8

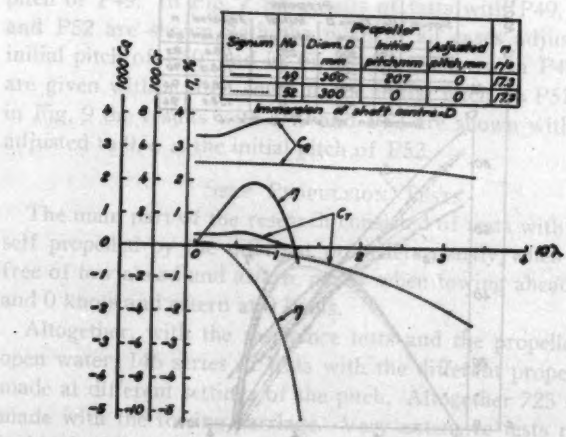

Propeller
Signum | No | Diam. D mm | Initial pitch, mm | Adjusted pitch, mm | n r/s
P49 | 300 | 207 | 0 | 12.3
P52 | 300 | 0 | 0 | 12.9
Immersion of shaft centre = D
1000 C_Q
100 C_T
η %
C_Q
η
C_T
η
10λ

Fig. 9

As already mentioned the available SHP at the shaft end was assumed to be 1200 at 154 Rpm. Similarly, the SHP was also assumed to be 800 at 140 Rpm. (This latter combination of power and revolutions was chosen so as to suit the propeller P49 with initial pitch when free of tow ahead. The fact that the diameter became somewhat too large for this case was, however, thereby disregarded.)

For investigating the influence of the revolutions, the SHP was, moreover, assumed to be 1200 at 231 Rpm. As already mentioned P82 was designed under this assumption.

In the comparison between propellers with fixed blades and propellers with adjustable blades it was assumed, for simplicity's sake that the range possessed by the engine is characterized by triangular diagrams according to Fig 10, *i. e.* the moment is constant and the revolutions limited by a governor. Also when going astern—either by reversing the direction of rotation of the shaft or with unaltered direction of rotation—it was assumed that the same engine characteristics apply.

Main Tests—The principal object of the investigation was to obtain, by means of a suitable adjustment of the pitch of the propellers, information on the following points, *viz*:

1.	Maximum speed when running free	At 1200 SHP, 154 Rpm. or 800 SHP, 140 Rpm. or in some cases at 1200 SHP, 231 Rpm.
2.	Maximum pull at 7 knots	
3.	Maximum pull at 3.5 knots	
4.	Maximum pull at zero speed	
5.	Maximum pull when going astern at zero speed	
6.	Maximum speed when running free astern	

All tests were conducted according to the "Continental" method[1]) with a tow rope force applied to the model equal to the skin friction correction for ship, reduced to model scale at each speed. The primary results with the model were transferred according to conventional methods, to the scale of the actual ship. All data in the following Figures and Tables refer to the actual ship. As the object of the investigations was, in the first place, to

[1] See: Congrès international des directeurs de bassins, Paris, 1935, p. 86.

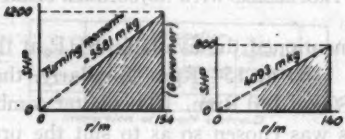


Fig. 10

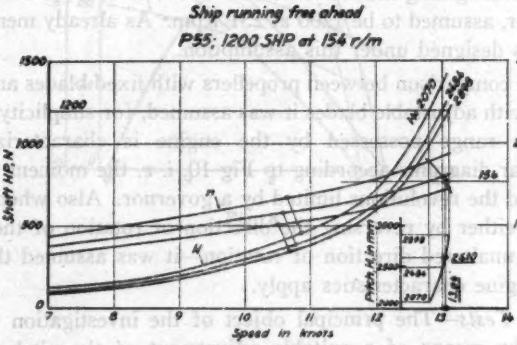


Fig. 11 a

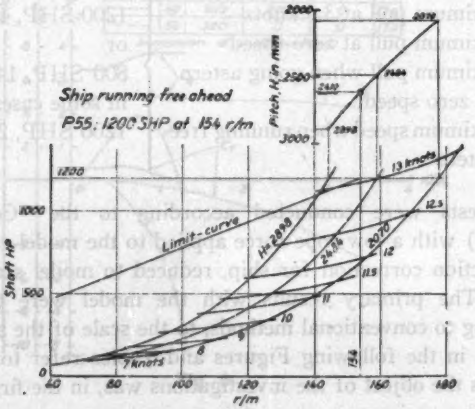


Fig. 11 b

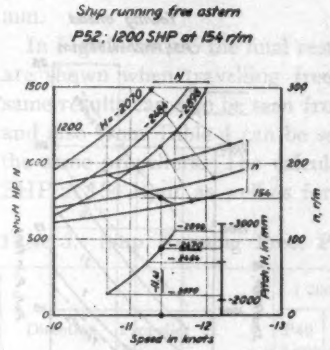


Fig. 12 a

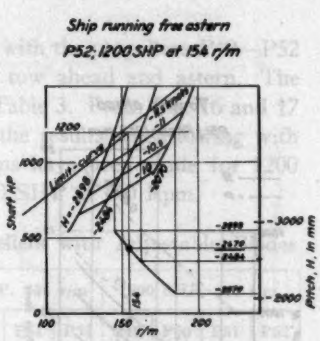


Fig. 12 b

make a *comparison* between different cases, no corrections have been made for roughness, scale effect, etc.

The tests when travelling free were made in the following way: With each propeller three (or more if required) self propulsion series were run with the blades set for different pitches. Fig. 11a and 11b give an example of how the results from these self propulsion series were worked out graphically. Fig. 11a shows how, by means of interpolation (or in some cases by extrapolation) the maximum speed and the correspondingly adjusted pitch is obtained. The way of representation according to Fig. 11b is a checking and at the same time a more careful determination of the adjusted pitch. The example represents the tests with propeller P55 when travelling free ahead. The propeller was then adjusted to pitches 2070, 2484, and 2898 mm. In Fig. 12a and 12b another example is shown applying to propeller P52 when going astern. The propeller was adjusted to the pitches — 2070, — 2484, and — 2898 mm.

The tests when towing were made in a similar way. Then arrangements were, moreover, made to measure the pull produced in each case. The pull was taken up by a horizontal wire 1.8 m above water level (at the ship's scale). For measuring the force in the wire the ordinary resistance dynamometer of the carriage was used. The method of working out the preliminary results can be seen from Fig. 13a, 13b, 13c, and 13d. Fig. 13d consists of cross curves at 154 Rpm. from the other figures. The example represents the tests with propeller P51 when towing ahead. The

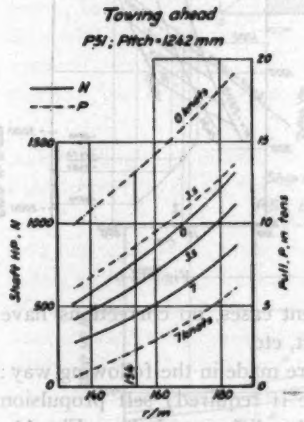


Fig. 13 a

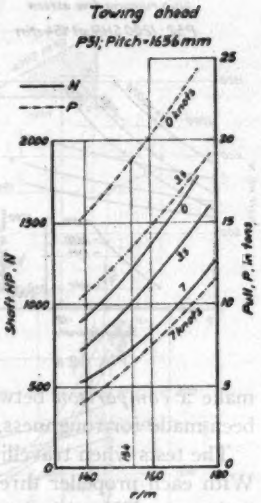


Fig. 13 b

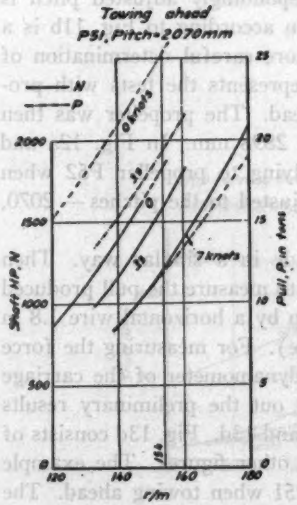


Fig. 13 c

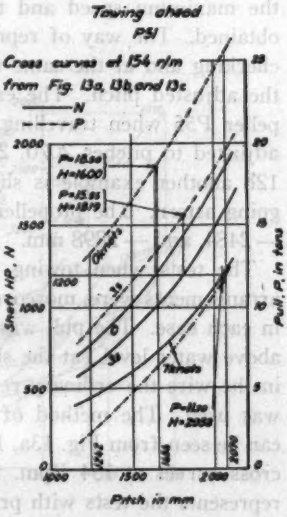


Fig. 13 d

propeller pitch was then set to the values 1242, 1656, and 2070 mm.

In Fig. 14 and 15 the final results with the propellers P49—P52 are shown when travelling free of tow ahead and astern. The same results can also be seen from Table 3. From Fig. 16 and 17 and also from Table 4 can be seen the results when towing with the same propellers. The calculations have been made for 1200 SHP at 154 Rpm. as well as for 800 SHP at 140 Rpm.

Table 3. Ship Running Free, Propellers with Adjustable Blades.

Direction	Propeller		1 200 SHP, 154 r/m				800 SHP, 140 r/m			
			P49	P50	P51	P52	P49	P50	P51	P52
	D	mm	3 600	3 600	3 600	3 600	3 600	3 600	3 600	3 600
	H_0	mm	2 484	1 656	828	0	2 484	1 656	828	0
Ahead	V_{max}	knots	12.91	12.86	12.83	12.75	12.31	12.21	12.15	12.10
	H	mm	2 484	2 550	2 555	2 490	2 445	2 500	2 484	2 465
Astern	$-V_{max}$	knots	10.48	10.79	11.12	11.41	9.20	9.56	9.82	10.25
	$-H$	mm	2 060	2 285	2 475	2 670	1 930	2 180	2 350	2 560

The results when going ahead have been plotted in Fig. 18 with the speed of the ship as abscissa.

From the results when travelling free ahead it can be seen that the initial pitch has a comparatively small influence on V_{max}. This

Table 4. Towing at Different Speeds, Propellers with Adjustable Blades.

Direction	Speed in knots	Propeller		1 200 SHP, 154 r/m				800 SHP, 140 r/m			
				P49	P50	P51	P52	P49	P50	P51	P52
		D	mm	3 600	3 600	3 600	3 600	3 600	3 600	3 600	3 600
		H_0	mm	2 484	1 656	828	0	2 484	1 656	828	0
Ahead	7	P	tons	11.95	11.75	11.70	11.20	7.95	7.90	7.80	7.60
		H	mm	2 010	2 070	2 058	2 020	1 940	2 015	1 995	1 965
Ahead	3.5	P	tons	16.20	16.10	15.55	15.05	11.70	11.55	11.30	10.85
		H	mm	1 750	1 810	1 819	1 780	1 632	1 700	1 730	1 675
Ahead	0	P	tons	19.35	19.20	18.50	17.70	14.55	14.30	14.00	13.25
		H	mm	1 600	1 645	1 650	1 590	1 475	1 520	1 540	1 480
Astern	0	$-P$	tons	7.30	8.60	9.35	10.20	5.20	6.20	6.70	7.30
		$-H$	mm	1 375	1 515	1 650	1 800	1 265	1 420	1 580	1 690

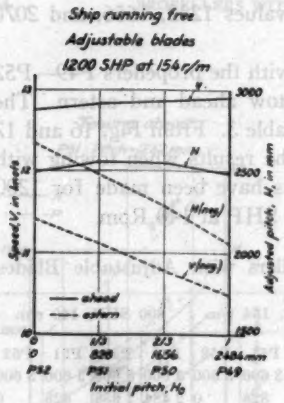


Fig. 14

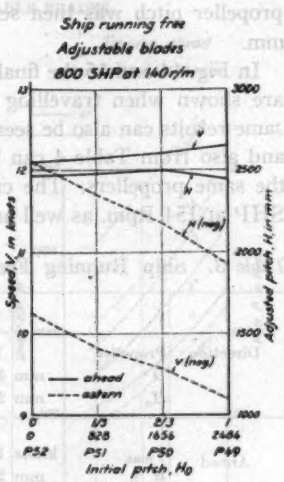


Fig. 15

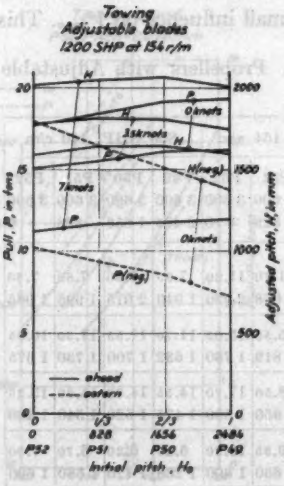


Fig. 16

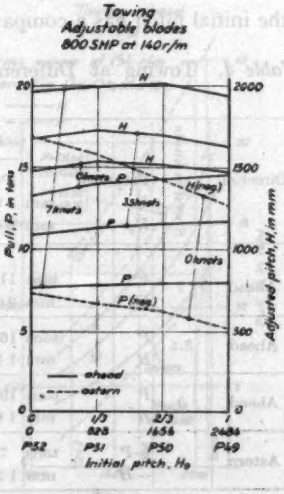


Fig. 17

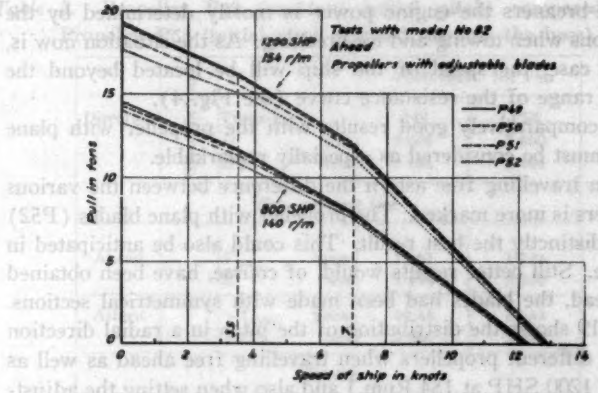

Tests with model No 82
Ahead
Propellers with adjustable blades
1200 SHP
154 r/m
800 SHP
140 r/m
P49
P50
P51
P52
Pull in tons
Speed of ship in knots

Fig. 18

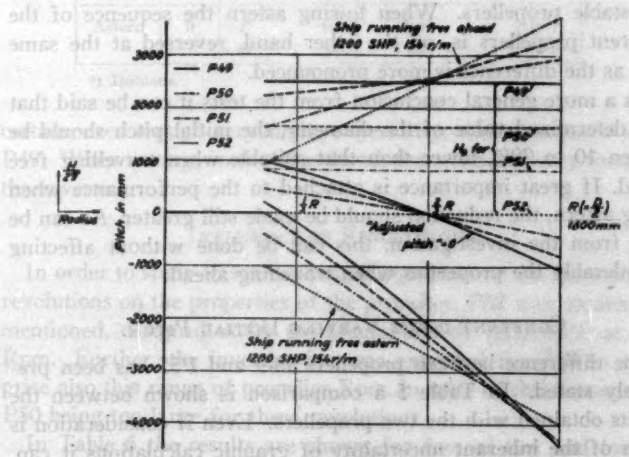

Ship running free ahead
1200 SHP, 154 r/m
P49
P50
P51
P52
Pitch in mm
Radius
"Adjusted pitch" = 0
Ship running free astern
1200 SHP, 154r/m

Fig. 19

might partly depend on the SHP 1200 and also 800 having to be considered as unnecessarily great for traveling free only. In tugs and ice-breakers the engine power is mostly determined by the conditions when towing and ice-breaking. As the situation now is, in this case, the speed of the ship will be located beyond the critical range of the resistance curve (see Fig. 4).

The comparatively good results with the propeller with plane blades must be considered as especially remarkable.

When travelling free astern the difference between the various propellers is more marked. The propeller with plane blades (P52) shows distinctly the best result. This could also be anticipated in advance. Still better results would, of course, have been obtained if, instead, the blades had been made with symmetrical sections.

Fig. 19 shows the distribution of the pitch in a radial direction for the different propellers when travelling free ahead as well as astern (1200 SHP at 154 Rpm.) and also when setting the adjustable pitch to zero.

From the results when towing ahead it can be seen that the highest values of pull are obtained with P49. The difference between P50 and P49 is, however, very slight. Nor is the difference between P52 and P49 considerable. In this circumstance there lies a certain possibility of standardizing the blades of adjustable propellers. When towing astern the sequence of the different propellers is, on the other hand, reversed at the same time as the difference is more pronounced.

As a more general conclusion from the tests it can be said that at a determined value of the diameter, the initial pitch should be chosen 10 to 20% lower than that suitable when travelling free ahead. If great importance is attached to the performance when going astern, the reduction should be made still greater. As can be seen from the investigation, this can be done without affecting considerably the properties when travelling ahead.

Constant *verses* Varying Initial Pitch.

The difference between propellers P49 and P55 has been previously stated. In Table 5 a comparison is shown between the results obtained with the two propellers. Even if consideration is taken of the inherant uncertainty of graphic calculations it can, however, be ascertained that, when travelling free, P55 shows a

Table 5. Propeller P49 (constant initial pitch) compared with Propeller P55 (initial pitch reduced towards the boss).

Direction	Speed in knots			1 200 SHP, 154 r/m	
		Propeller		P49	P55
		D	mm	3 600	3 600
		H_0	mm	2 484	2 484
Ship running free					
Ahead		V_{max}	knots	12.91	13.04
		H	mm	2 484	2 610
Astern		$-V_{max}$	knots	10.48	10.54
		$-H$	mm	2 060	2 020
Towing					
Ahead	7	P	tons	11.95	12[1])
		H	mm	2 010	2 020[1])
Ahead	3.5	P	tons	16.20	16.10
		H	mm	1 750	1 820
Ahead	0	P	tons	19.35	19.0
		H	mm	1 600	1 680
Astern	0	$-P$	tons	7.30	7.30[1])
		$-H$	mm	1 375	1 320[1])

[1]) Dubious.

certain, even though very slight, superiority, when compared with P49. When towing, on the other hand, there is a tendency towards the reverse.

Influence of Revolutions.

In order to study to some extent the question of the influence of revolutions on the properties of the propeller, P82 was, as already mentioned, designed under the assumption of 1200 SHP at 231 Rpm. Further, the investigation was extended for P50 to comprise also this range of propeller Rpm. in spite of the diameter of P50 being too large for these revolutions.

In Table 6 the results are shown for free of tow and towing with P82 and P50 under this assumption. At the same time from Tables 3 and 4 are given the results with P50, P49 and P52 under the assumption of 1200 SHP at 154 Rpm. P50 has been included

Table 6. Influence of Revolutions.

Direction	Speed in knots			1 200 SHP, 231 r/m		1 200 SHP, 154 r/m		
		Propeller		P82	P50	P50	P49	P52
		D	mm	3 120	3 600	3 600	3 600	3 600
		H_0	mm	1 440	1 656	1 656	2 484	0
Ship running free								
Ahead		V_{max}	knots	12.92	11.67	12.86	12.91	12.75
		H	mm	1 610	1 160	2 550	2 484	2 490
Astern		$-V_{max}$	knots	10.48	—	10.79	10.48	11.41
		$-H$	mm	1 475	—	2 285	2 060	2 670
Towing								
Ahead	7	P	tons	10.15	5.60	11.75	11.95	11.20
		H	mm	1 274	820	2 070	2 010	2 020
Ahead	3.5	P	tons	14.0	9.30	16.10	16.20	15.05
		H	mm	1 166	635	1 810	1 750	1 780
Ahead	0	P	tons	16.75	12.0	19.20	19.35	17.70
		H	mm	1 084	555	1 645	1 600	1 590
Astern	0	$-P$	tons	6.30	3.10[3])	8.60	7.30	10.20
		$-H$	mm	1 035	600[3])	1 515	1 375	1 800

[3]) Dubious

for comparison with P50 at 231 Rpm. P49 represents the best results when free of tow and towing ahead, whereas P52 represents the best result when going astern.

As can be seen in a comparison between P82 (231 Rpm.) and P49 (154 Rpm.) the results when free of tow (ahead as well as astern) are almost identical. These results must be considered as unexpected. According to the investigation in appendix 1 the probable efficiency (in open water) for P82 (231 Rpm.) might be expected to be considerably lower than that of P49 (154 Rpm.). The reason might be partly that the speeds fall beyond the critical range of the ship's resistance curve. When towing (ahead as well as astern) P49 is, on the other hand, clearly superior to P82, *i. e.* the performance when towing is considerably greater at the lower Rpm.

In a comparison between P82 and P50, both at 231 Rpm. it is clear that the results with P50 are extremely bad. When travelling

free this is due to the diameter of P50 being too large. This applies however, also when towing. As can be seen from the investigation in appendix 2 the performance, when towing at the values of pitches which are used for P50, is very poor. A comparison between P50 (231 Rpm.) and P50 (154 Rpm.) is of interest in this connection.

Although the investigation is incomplete it proves the importance of keeping the revolution relatively low in order to obtain good towing properties. This result might also have been expected from the very beginning.

Fixed Blades *verses* Adjustable Blades.

The advantage of propellers with adjustable blades as compared with propellers with fixed blades is most pronounced in such cases where the propeller has to work under very varying load conditions as for tugs and ice-breakers. The reason is, of course, that with adjustable blades the entire SHP can be utilized in any situation, which is not the case with fixed blades. If the propeller with fixed blades is designed for free of tow it will be "too large" when towing. If, on the other hand, the propeller is designed for towing it becomes "too easy" when travelling free.

To make this matter clear in the present case the propellers P49 and P50 with fixed initial pitch are used as objects of experiment. The initial pitch for P49 was calculated for travelling free, whereas the initial pitch for P50, in the main, was suited for heavy towing (at zero speed). The initial pitch of these propellers thus represented the extreme cases. In practical cases a compromise between these extreme cases is often chosen. As an example of such a compromise P49 is used with the pitch set at the value 2070 mm. (In this investigation no consideration was taken of the shape of the boss not being that which is generally used with fixed blades.)

Ahead—Fig. 20 shows results with P49 with fixed (initial) pitch and from Fig. 21 the results can be seen with P50 with fixed (initial) pitch. In Fig. 22 is shown the case of compromise with P49 with the pitch set at 2070 mm. In all cases it is assumed that the data of the engine are 1200 SHP at 154 Rpm. In Fig. 21 and 22 it is, as can be seen, assumed that the revolutions are limited by a governor. If this is not the case and a higher speed

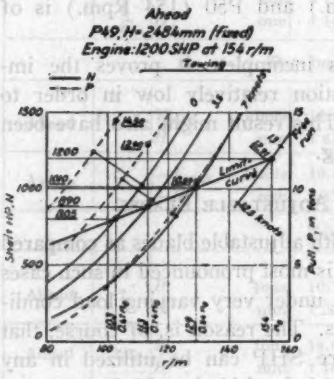


Fig. 20

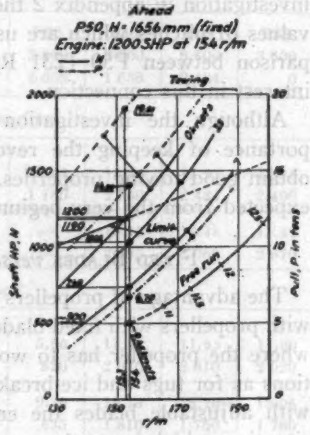


Fig. 21

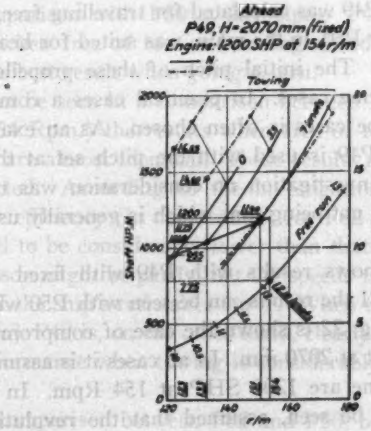


Fig. 22

Table 7. Ship Running Free, Ahead and Astern, Adjustable Blades *versus* Fixed Blades.

Direction	Propeller		1 200 SHP, 154 r/m		800 SHP, 140 r/m	
			P49	P50	P49	P50
	D	mm	3 600	3 600	3 600	3 600
	H_0	mm	2 484	1 656	2 484	1 656
Adjustable blades						
Ahead	V_{max}	knots	12.91	12.86	12.31	12.21
	H	mm	2 484	2 550	2 445	2 500
Astern	$-V_{max}$	knots	10.48	10.79	9.20	9.56
	$-H$	mm	2 060	2 285	1 930	2 180
Fixed blades (Pitch = H_0)						
Ahead	V_{max}	knots	12.91	9.45[1])	12.29	8.7[1])
	N	SHP	1 200	500[1])	790	375[1])
	n	r/m	154	154	138.5	140
Astern by reversing	$-V_{max}$	knots	10.93	7.40	10.05	6.8
	N	SHP	845	400	620	305
	$-n$	r/m	154	154	140	140

[1]) Dubious.

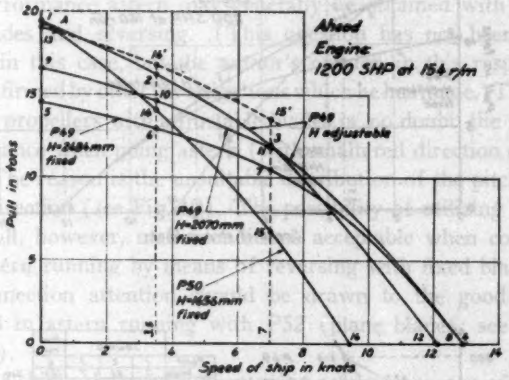
is permissible (as when steam reciprocating engines are concerned) the situation is, of course, somewhat improved. In Tables 7 and 8 the same results are shown (but not those from Fig. 22), also the results from similar graphical calculations made under the assumption 800 SHP at 140 Rpm. At the same time, for comparison, the results with adjustable blades have been given from Tables 3 and 4.

In Fig. 23 and 24 the results obtained have been plotted with the ship speed as abscissa.

Astern—In the case of propellers with adjustable blades, astern running is effected, with unaltered direction of rotation of the shaft, by turning the blades into the astern position. The fact that the direction of rotation does not have to be reversed might involve a simplification and thus lower the price of the engine. With fixed blades, astern running must, of course, be brought about by reversing the direction of rotation of the propeller shaft. Of course, with adjustable blades astern running can also be obtained by reversing, in which case the same setting of the

Table 8. **Towing Ahead and Astern, Adjustable Blades *versus* Fixed Blades.**

Direction	Speed in knots		1 200 SHP, 154 r/m		800 SHP, 140 r/m	
		Propeller	P49	P50	P49	P50
		D mm	3 600	3 600	3 600	3 600
		H_0 mm	2 484	1 656	2 484	1 656
Adjustable blades						
Ahead	7	P tons	11.95	11.75	7.95	7.90
		H mm	2 010	2 070	1 940	2 015
Ahead	3.5	P tons	16.20	16.10	11.70	11.55
		H mm	1 750	1 810	1 632	1 700
Ahead	0	P tons	19.35	19.20	14.55	14.30
		H mm	1 600	1 645	1 475	1 520
Astern	0	$-P$ tons	7.30	8.60	5.20	6.20
		$-H$ mm	1 375	1 515	1 265	1 420
Fixed blades (Pitch = H_0)						
Ahead	7	P tons	10.50	6.70	7.05	4.40
		N SHP	1 010	740	660	515
		n r/m	129	154	116	140
Ahead	3.5	P tons	12.90	14.00	9.00	11.10
		N SHP	890	1 025	560	760
		n r/m	113	154	98.5	140
Ahead	0	P tons	14.50	19.01	10.45	13.80
		N SHP	805	1 190	500	750
		n r/m	103	153	87.5	131
Astern by reversing	0	$-P$ tons	9.50	8.90	7.35	7.40
		N SHP	880	920	550	695
		$-n$ r/m	113	154	97	140


Ahead
Engine
1200 SHP at 154 r/m
P49
H adjustable
P49
H=2484mm
fixed
P49
H=2070mm
fixed
P50
H=1656mm
fixed
Pull in tons
Speed of ship in knots

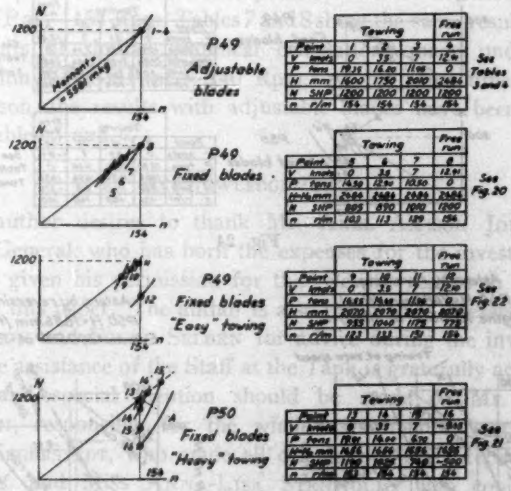
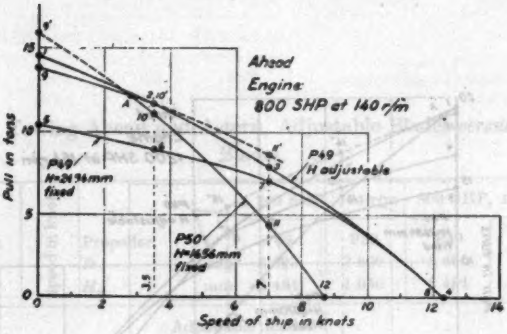

P49
Adjustable
blades
Towing
Free run
See Tables 3 and 4
P49
Fixed blades
See Fig. 20
P49
Fixed blades
"Easy" towing
See Fig. 22
P50
Fixed blades
"Heavy" towing
See Fig. 21

Fig. 23



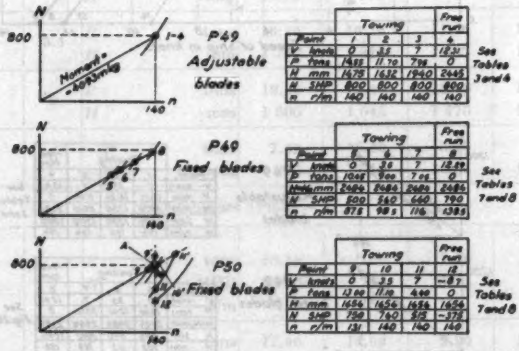


Fig. 24

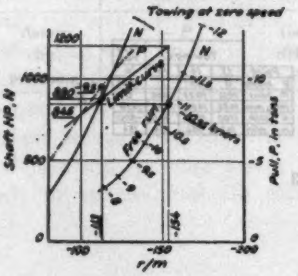


Fig. 25

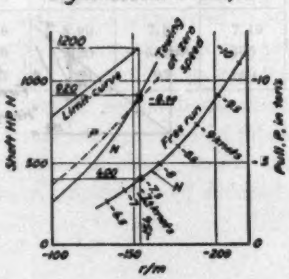


Fig. 26

blades as when cruising ahead is maintained. Then the setting of the blades can be adjusted so that full SHP is utilized in any situation. This is generally not the case with fixed blades, when the pitch is designed either for travelling free or for towing. The best performance astern may generally be obtained with adjustable blades and reversing. (This question has not been investigated in this case, but the author's opinion in this respect has been confirmed by other investigations which he has made.) The weak point in propellers with adjustable blades is, no doubt, the reduced performance when going astern (with unaltered direction of rotation). The reason is the unsuitable distribution of the pitch in the radial direction (see Fig. 19). The possibility of utilizing the full SHP will, however, make conditions acceptable when compared with astern running by means of reversing with fixed blades. In this connection attention should be drawn to the good results obtained in astern running with P52 (plane blades; see Tables 3 and 4).

Fig. 25 shows the results when going astern by means of reversing with P49 and Fig. 26 shows those with P50, in both cases with the pitch set at the initial pitch and under the assumption 1200 SHP at — 154 Rpm. Tables 7 and 8 show the same results, also the results of similar graphical calculations made under the assumption 800 SHP at — 140 Rpm. At the same time, for a comparison, the results with adjustable blades have been given from Tables 3 and 4.

ACKNOWLEDGEMENT.

The author desires to thank Mr. AXEL AX:SON JOHNSON, Consul General, who has born the expenses for the investigation and has given his permission for the relevant results to be collected in this paper. The author is also indebted to Messrs, ELOV ENGLESSON and FOLKE SELDÉN for advice during the investigation. The assistance of the Staff at the Tank is gratefully acknowledged, and special mention should be made of Mr. ARNE HANSSON, responsible for the whole experimental work, Mr. EVERT SEGERSTEDT, who made all calculations and prepared the diagrams, and Miss ANNA-LISA STENHOLM, who copied the diagrams.

Appendix 1.

Calculation of the Propeller Dimensions When Free of Tow.

To get a survey of the principal propeller data which might be used if the propeller is designed for free running only, calculations were made using the propeller diagram B. 3.50[1]) published by Troost. The calculations were based on the following assumptions:

1200 SHP

154 Rpm. and 231 Rpm. respectively

Speed of advance in both cases = 8.5 knots, corresponding to a supposed wake fraction w (Taylor) of about 0.35 ($V_A \sim 13$ knots)

The calculations were made for the following values of the pitch ratio: 0.6, 0.7, 0.8, 0.9, and 1.0. The results of the calculations can be seen from Fig. 27. The maximum efficiency for 154 Rpm. is located at a pitch ratio

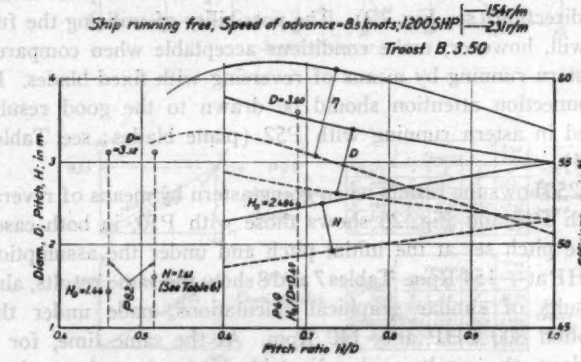


Fig. 27

of about 0.65 and amounts to about 61% (in open water). The dimensions chosen for P49 have been drawn in the figure and are in quite good conformity with the calculated curves. For 231 Rpm. the maximum efficiency seems to be located at H/D about 0.57. For P82 was chosen $D = 3.12$ m and $H_0 = 1.44$ m, *i. e.* $H_0/D \sim 0.46$. From reasons already indicated the diameter was chosen somewhat too large. From Table 6 it can be seen that when travelling free ahead the pitch must be adjusted to the value 1.61 m, *i. e.* H/D about 0.52. According to Fig. 27 the probable value of the efficiency then will be about 56%. There is every possibility that the results when free of tow would have been better with P49 (154 Rpm.) than with P82 (231 Rpm). As already stated this, however, did not prove to be the case.

Appendix 2.

Calculation of Propeller Dimensions for "Heavy" Towing.

It might be of interest for the present investigation to form an idea of the propeller dimensions, which would be used if the propeller were designed only to give the greatest pull for heavy towing (at zero speed). To make such an investigation the results known by the author, which

[1] Trans. N. E. Coast Inst. of Eng. & Shipb., Vol. LVI, p. 91. See also: W. P. A. van Lammeren, Weerstand en Voortstuwing van Schepen, Amsterdam, 1942, p. 197.

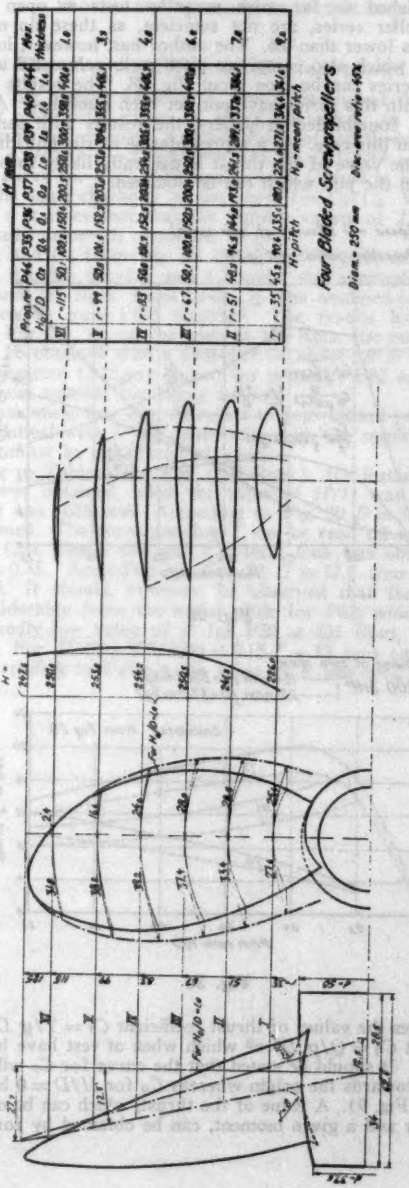

Four Bladed Screwpropellers
diam = 250 mm
H = pitch
H_m = mean pitch

Fig. 28

have been published so far, with propeller tests in open water with systematic propeller series, are not sufficient, as these do not comprise pitch ratio values lower than 0.6. The author has, however, investigated a propeller series, which also comprises pitch ratio values 0.2 and 0.4. The details of this series can be seen from Fig. 28. The results of the tests in open water with this series have not yet been published. Although the series comprises four-bladed propellers, the results with same will give some hints also in this case. As a representative of the pull the thrust will then be used. The value of the thrust is generally likely to be about 5 to 10% greater than the pull which can be obtained.

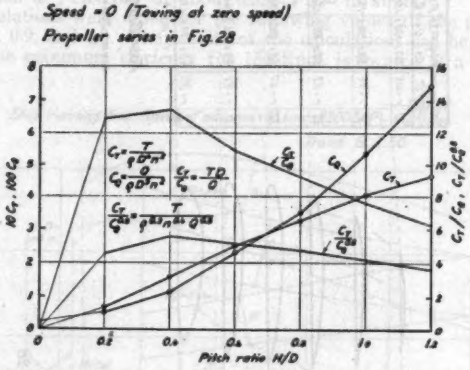


Fig. 29

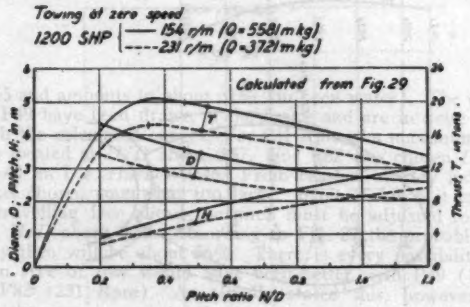


Fig. 30

Fig. 29 indicates the values of thrust coefficient $C_T = T/\varrho\, D^4\, n^2$ and the torque coefficient $C_Q = Q/\varrho\, D^5\, n^2$ which when at rest have been obtained with this series. It should be stated that the curve for C_T will, practically speaking, go in towards the origin whereas C_Q for $H/D = 0$ has a positive value (compare Fig. 9). A value of the thrust, which can be obtained with a given diameter and a given moment, can be obtained by forming C_T/C_Q

$\left(= \frac{TD}{Q}\right)$. The curve for C_T/C_Q has also been plotted in the figure. As can be seen the highest value of T (and thus also of the pull P) can be expected at pitch ratio values of about 0.3 to 0.4. At lower pitch ratio values T drops enormously. Another value of the thrust can be obtained by forming $C_T/C_Q^{0.8}$ $(= T/\varrho^{0.2} n^{0.4} Q^{0.8})$. In $C_T/C_Q^{0.8}$ n comes in instead of D. The curve for $C_T/C_Q^{0.8}$ has also been plotted in Fig. 29. It is, as a whole, of the same character as the curve for C_T/C_Q. With given values for Q and n it is evident that the highest values of T can be expected for pitch ratio values of about 0.3 to 0.4.

Using C_T and C_Q from Fig. 29 D, H, and T have been calculated for $H/D = 0.2$, 0.4, 0.6, 0.8, 1.0, and 1.2 under the assumption 1200 SHP at 154 Rpm. and 231 Rpm. respectively. ϱ was assumed $= 103$ kg sec^2/m^4, corresponding to $\varrho g = 1.015$ tons/m^3. The results have been brought together in Fig. 30. As can be seen, at 154 Rpm. the maximum thrust at rest should be obtained with a propeller of about 3.9 m diameter. At 231 Rpm. the diameter (3.12 m) chosen for propeller P82 seems, as a whole, to be the most suitable for heavy towing.

When considering that Fig. 30 refers to four-bladed propellers and that T is not identical with P the conformity with the results obtained in the model tests must be considered as good.

According to Table 6 for P50 (154 Rpm.), for instance, the pull $P =$ 19.20 tons was obtained, when the value of H/D was $1645/3600 \sim 0.46$. H_0 for P50 was 1656 mm. According to Fig. 30 $D = 3.6$ m for $H/D =$ 0.46 is obtained. The corresponding T can be read off as being 19.9 tons.

For P82 (231 Rpm.) the pull $P = 16.75$ tons was obtained at $H/D =$ $1084/3120 = 0.35$. According to Fig 30 $T = 17.2$ tons is obtained for $H/D = 0.35$. It should, however, be observed that the pitch 1084 mm differs considerably from the initial pitch for P82, which was 1440 mm.

The markedly low value of P for P50 at 231 Rpm. is also explained by Fig. 30. For $H/D = 555/3600 = 0.15$ $T \sim 13$ tons can be read off in Fig. 30. According to Table 6 the value of P was $= 12$ tons.

STABILITY OF SHIPS AFTER EXTENSIVE FLOODING.

By Charles L. Wright, Jr.*

Introduction.

Marco Polo, in an account of his travels through the East during the latter part of the eleventh century, described the merchant vessels of that time and noted:

> "Some ships of the larger class have as many as thirteen bulkheads or divisions in the hold - - - . The object of these is to guard against accidents which may occasion the vessel to spring a leak, such as striking on a rock or receiving a stroke from a whale - - - . The water, running in at the place where the injury has been sustained, - - - cannot pass from one division to another."

Apparently this protection of ships against sinking due to underwater damage was a matter of greater concern to the medieval seafaring people of the East than it is to some builders and operators of today. Many vessels now in service could not survive such damage and many more could not survive if the damage involved one of the few subdivision bulkheads with which they are provided.

A textbook on "Hydraulics and Mechanics", published more than a hundred years ago, contained a discussion on the reluctance of American and European nations to adopt a system of subdividing the holds of their ships similar to the system described by Marco Polo. This discussion included the following paragraph which pretty well expresses the present situation:

> " - - - beside the opposition of popular prejudice and the increased expense, an objection might arise from the reduction it would occasion in the quantity of freight, and the increased difficulty of stowing bulky articles. It remains to be considered how far these objections ought to prevail against the greater security of the vessel, crew and cargo. At any rate, such objections do not apply to ships of war, in which to carry very heavy burdens, is not an object of consideration."

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The first legal requirements for watertight bulkheads in merchant vessels were put into effect by the British Marine Shipping Act of 1854, and since that time an increasing amount of subdivision has been required by classification society regulations and international law. The subdivision of merchant vessels is now largely determined by the "Convention, with Regulations," prepared at the international Conferences on Safety of Life at Sea, 1929.

It may appear that if any more bulkheads were installed in a ship than the minimum required by the Convention, the ship would be seriously handicapped in competition for international trade. However, the U. S. Maritime Commission has designed and built many ships which clearly demonstrate that improvements, not only in subdivision but also in stability after flooding, are practicable and can be economically attained. In any event, it is axiomatic that whatever bulkheads are installed should be so arranged as to provide the greatest ability to survive underwater damage for the least encumbrance upon the usefulness and operation of the vessel.

In time of war, most merchant vessels are pressed into military service and hence are subjected to a much greater danger of underwater damage than in normal peacetime operation. If subdivision studies are made during the design of a vessel, an arrangement can be developed such that the bulkheads installed during construction will provide the desired ability to survive the normal hazards of the sea, and additional bulkheads contemplated to be installed in time of emergency will improve the ability to survive to that considered satisfactory for naval auxiliaries.

As to what ability to survive is satisfactory for naval auxiliaries or for combatant vessels, and as to what inconvenience in loading and operating should be accepted are matters of judgment largely beyond the scope of this atricle. Prior to the outbreak of World War II, troop transports were considered satisfactory if they met the two compartment standard of subdivision required of large passenger vessels. It was in connection with the conversion of passenger vessels to troop transports that the inadequacy of the two compartment standard was fully realized. No very extensive damage was necessary to open more than two compartments, especially if short compartments were accepted.

In connection with the conversion of merchant vessels and the

improvement of combatant ships during World War II, the subdivision and stability of over a hundred classes of vessels were studied in detail. The methods previously in use for calculating and presenting the information necessary to determine the ability of vessels to survive damage were soon found to be far from adequate. New methods were developed and a series of model tests was conducted to check the applicability of the new methods to each special type of problem. The principles of these methods, together with an explanation of their advantages over previous methods, and their need, are presented herein for consideration in the design of all vessels, both merchant and military.

Certain steps that still follow previously accepted procedures are not described in detail but are covered by references to standard text books.

Analysis of Ability to Survive.

The extent of underwater damage a vessel can survive depends upon:

(*a*) The arrangement of subdivision bulkheads with respect to the floodable length of the vessel.

(*b*) The residual stability of the intact portions of the vessel after the flooding caused by the damage.

The steps involved in the complete study of any design include:

(*a*) Preparation of a subdivision plan upon which the arrangement of all watertight compartmentation is developed, and corresponding floodable length curves are shown.

(*b*) Careful study of this plan to determine the extent of damage the vessel can sustain before the resulting flooding exceeds the floodable length.

(*c*) Further study to determine whether this extent of damage can be increased by adding or relocating subdivision bulkheads or by establishing a maximum allowable operating draft without seriously limiting the usefulness of the ship.

(*d*) Calculation of residual righting arms after each potentially critical case of flooding that could result from the extent of damage determined, and based on several arbitrary conditions of loading spanning the operating range of the vessel.

(*e*) Calculation of heeling arms that could result from unsymmetrical flooding of damaged compartments and empty tanks.

(*f*) Preparation of stability curves, plotted from the results of the above calculations with curves of heeling arms superimposed on curves of righting arms, showing the initial list and residual stability after each case of flooding for each arbitrary condition of loading.

(*g*) Determination of actual conditions of loading and interpolation of the stability curves for the arbitrary conditions of loading to give similar stability curves for the actual conditions.

(*h*) Careful study of the most critical of these curves to determine any improvements that might be effected by changes in weight or weight distribution including solid ballast.

(*i*) Preparation of a procedure for the proper loading of the vessel and for the employment of liquids and liquid ballast to provide as much stability as possible in case of damage, and to reduce the heeling moment that might occur due to unsymmetrical flooding.

Stability Based on Subdivision.

The subdivision plan shown on Plate I was prepared for the studies of a class of troop transports converted from cargo vessels. Main transverse bulkheads shown by heavy solid lines were included in the original design. Bulkheads shown by heavy dotted lines were added as a result of these studies. Inner bottom plating, wing tanks, deep tanks, and all other watertight subdivisions that could form intact spaces in flooded compartments are indicated in plan and profile. Plan views at more than one level are sometimes needed, but usually one plan view through the hold is sufficient. The use of each compartment is shown to aid in estimating the permeability (or per cent of volume floodable). Floodable length curves are plotted over the profile to indicate the groups of adjacent compartments that can be flooded without submerging the "margin line."

The residual stability of the portions of the hull remaining intact after the flooding of any group of compartments, is an even more important consideration than the fact that the compartments can be flooded without submerging the "margin line."

A vessel with insufficient stability may not remain upright nearly as long as a vessel with insufficient floodable length will stay afloat. Ships have been reported to capsize and sink in less than ten minutes after damage with accompanying large loss of life. On the other hand, the U.S.S. *Susan B. Anthony*, a troop transport, formerly the passenger vessel *Santa Clara*, while participating in the invasion of the Normandy Coast, June 7, 1944, received damage that resulted in flooding considerably more than her floodable length; yet this vessel had sufficient stability to remain upright, and consequently stayed afloat for two hours and fifteen minutes while all on board were rescued.

A vessel which does not actually capsize but which develops a large angle of heel due to negative initial stability or excessive unsymmetrical flooding, is little better than lost as the execution of salvage and abandon ship operations is rendered practically impossible. On November 12, 1942, the U.S.S. *Tasker H. Bliss*, a troop transport, formerly the passenger vessel *President Cleveland*, was torpedoed while unloading at a port of debarkation on the African Coast. The vessel heeled to an angle of 30° due to insufficient stability after damage, and although she remained afloat for nearly nine hours, salvage operations were of no avail.

In order to assure sufficient stability in case of damage, it is often necessary to provide more stability for the intact ship than would otherwise be desired. Consideration should be given to the danger of a short period of roll synchronizing with the period of the waves. Ability to survive damage warrants a sacrifice in comfort, but it does not warrant any risk to the intact vessel. Since results must indicate excessive as well as insufficient stability, no greater margin for error should be allowed in one direction than in the other. Assumptions should be based on average conditions and calculations should be as accurate as the assumptions warrant.

Stability after extensive flooding is the same in principle as stability before damage. The buoyant volume becomes the volume displaced by the intact portions of the vessel and the impermeable material that remains within the flooded compartments. The accuracy with which the amount and arrangement of the impermeable material in the flooded compartments can be estimated is an important factor in the accuracy of the final results. If the same values are consistently assigned to similar spaces in different studies, results will be at least comparative.

Flooding usually causes a change in trim and a change in the angle of inclination as well as causing sinkage to a deeper waterline. Trim causes additional flooding. A change in the angle of inclination may cause an additional change in trim. Trim, heel, sinkage, and flooding all affect each other, and each affects the stability after damage.

The significance of each of these variables should be carefully considered before the involved calculations necessary to determine the effect of the variable are undertaken. Any variable that is less significant than the inevitable inaccuracies in estimating permeabilities may be neglected. Other variables should be considered with respect to their significance as compared to the time required to calculate their effect. In the studies made during World War II, it was considered more important to obtain a fair indication of the stability of as many ships as possible, than to take time for better accuracy with the consequent reduction in the number of ships that could be studied.

Accordingly, trim was considered to be a factor that could be neglected in all studies. This course of action was consistent with practically all studies ever made of stability before damage, as few cross curves of stability have been based on trimmed waterlines.

Most critical cases of flooding are those in which the compartments damaged are located near midships, and little trim results from flooding in this vicinity. A comparison of model tests based on parallel sinkage with those based on normal trim shows that the effect of trim is in no case appreciable, as will be described.

Presentation of Stability Data.

Stability is the tendency of a vessel, when inclined to an angle, to return to its initial upright position. The best mathematical representation of this tendency is the righting moment of the vessel. A righting moment is not an easy value to visualize, but when divided by the displacement of the vessel, it becomes a righting arm and can then be conceived as a linear dimension.

Righting arm values (GZ) for angles of inclination (θ) approaching zero, can be expressed as a function of the initial transverse metacentric height (GM).

$$GZ = GM \sin \theta$$

Prior to World War II metacentric heights rather than righting arms were used almost universally as the means of expressing stability after flooding. This was probably due to the fact that the relation of the position of the metacenter to the moment of inertia of the waterplane makes it much simpler to determine metacentric heights than to calculate righting arms.

Metacentric heights do serve as an indication of the period of roll of the vessel and they show whether the vessel will take an initial list due to negative stability, but they give no indication whatsoever of stability at any appreciable angle of inclination.

The unsymmetrical flooding that may occur after extensive damage often results in an appreciable heeling moment. When righting arms rather than metacentric heights are used, this heeling moment can be expressed as a heeling arm and compared directly with the righting arm to determine the angle of heel and the residual stability.

The metacentric heights may be calculated in addition to the righting arms, so that "GM tangent lines" can be drawn.* The righting arm curves (or curves of statical stability) can then be faired into the tangent lines at 0°. This procedure aids in determining the shape of the righting arm curves at small angles of inclination where righting arm values cannot be as accurately calculated.

The many different studies made during World War II indicate both before and after damage that some vessels with large metacentric heights have such small righting arms at any appreciable angle of inclination that they would be quite apt to capsize without warning, whereas other vessels with negative metacentric heights have such good righting arms beyond the angle of initial list that there would be no danger of capsizing at all. These studies also indicate that for some vessels at light drafts the loss in metacentric height due to flooding is large, while the loss in righting arms is not particularly appreciable; on the other hand, at deep drafts the change in metacentric height is a gain rather than a loss, while the loss in righting arms at large angles of inclination is very pronounced. It is becoming quite generally agreed that righting arm values are essential in presenting a complete indication of the stability characteristics of a vessel.

For any specific case of damage in any specific condition of

*Rossell, Henry E. and Chapman, Lawrence B. "Principles of Naval Architecture," Volume I, Page 120, the Society of Naval Architects and Marine Engineers, 1939.

loading, the stability before and after flooding can be clearly represented by a plot such as that illustrated in Figure 1. The following pertinent information should be noted:

(*a*) Condition of loading: draft (d_0), displacement (Δ_0), and height (KG) of the center of gravity above the base before damage.

(*b*) Extent of the damage.

(*c*) Arrangement of impermeable material as assumed.

The metacentric height before damage (GM_0) and after damage (GM_τ) are plotted at 57.3° (1 radian), and the "GM tangent lines" are drawn. Righting arms before damage (GZ_0) and after damage (GZ_τ) are plotted, and curves of statical stability are drawn. Heeling arms (Oz) due to unsymmetrical flooding are plotted and the curve of heeling arms is drawn.

The lost GM and the loss in righting arms due to flooding are then clearly evident. In the case of negative initial stability, the intersection of the righting arm curve with the base line indicates the initial list. In the case of unsymmetrical flooding, the intersection of the residual righting arm curve with the heeling arm curve indicates the initial angle of heel. The difference between the residual righting arm curve and the heeling arm curve at any angle is the actual righting arm at that angle, and the area between the two curves up to any angle is a measure of the dynamic stability.

Although some positive righting arm or dynamic stability is necessary to prevent wind, waves, and other external forces from capsizing the vessel, or heeling the vessel to an angle that causes flooding through openings in the intact portion of the hull, it cannot be expected that a damaged ship will be able to survive heavy rolling. Righting arm curves after damage need seldom be carried beyond 50°.

Procedure for Calculations.

Every ship presents individual problems in determining stability characteristics. Each case of damage, each condition of loading, requires detailed analysis of all significant factors; and care must be taken that any unusual factor is not overlooked. It is not possible to prepare detailed instructions that would apply to all vessels. The procedure to be described is intended only as an example of the type of studies that should be made and the factors that should be considered.

CONDITION OF LOADING:

d_o, Δ_o, KG

CARGO - AMOUNT, TYPE AND LOCATION

LIQUIDS - LOADING OF TANKS

CASE OF FLOODING:

EXTENT OF DAMAGE

DAMAGED TANKS

INTACT SPACES

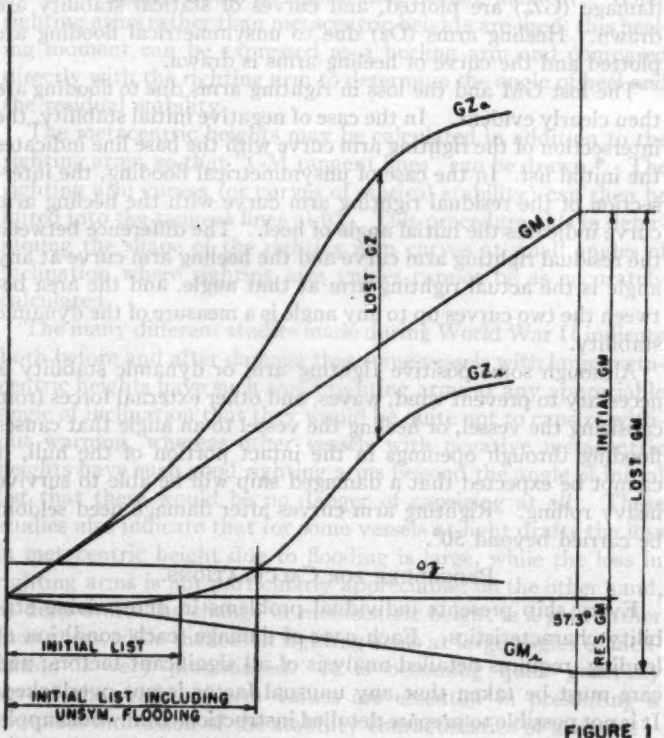


FIGURE 1

War experiences of the vessels which were studied in accordance with this procedure, and model tests, performed to check the accuracy of the procedure, show that a reliable indication of stability characteristics can be obtained, provided that the vessels are of moderate hull form and the impermeable material in flooded compartments can be reasonably well represented. For vessels of extreme hull form, or for cases of damage near the bow or stern, satisfactory accuracy may require that a model of the particular ship be constructed and that the stability characteristics be obtained by actual tests. Models for stability tests of several different types of combatant vessels are now being constructed by the Navy Department.

Direct calculations for the stability characteristics of an undamaged vessel can be prepared by any of several well established methods.* The righting arm values can be plotted as cross curves of stability from which the values corresponding to any condition of loading may be quickly determined.

Calculation for stability characteristics after extensive flooding involve the following considerations:

(*a*) The flooded portions of the damaged compartments no longer form a part of the buoyant volume of the ship. This loss in buoyancy causes the vessel to sink to a deeper waterline.

(*b*) Unless material is torn away by the damage, the weight (Δ_0) of the vessel and the height (KG) of the center of gravity above the keel remain the same as before the damage occurred.

(*c*) The water displaced by the intact portions of the vessel at the deeper waterline is the same in weight as that displaced by the intact ship before damage, but the volume of displacement is different in shape hence causing the differences between the stability characteristics before and after damage.

A procedure could be employed similar to that for calculating the stability characteristics of an intact vessel, using stations taken through the intact portions of the vessel and the impermeable material within the flooded compartments. However, where calculations for the intact vessel have already been made, the work involved can be considerably reduced by calculating the

*Ibid pages 105-107, 117, 118.

loss of stability in the flooded compartments and deducting this loss from the stability of the intact vessel. The waterlines used for the calculations of the intact vessel must cover sufficient range so that values can be obtained for the deep waterlines to which the vessel may sink after flooding. This range may extend practically to the "bulkhead deck."

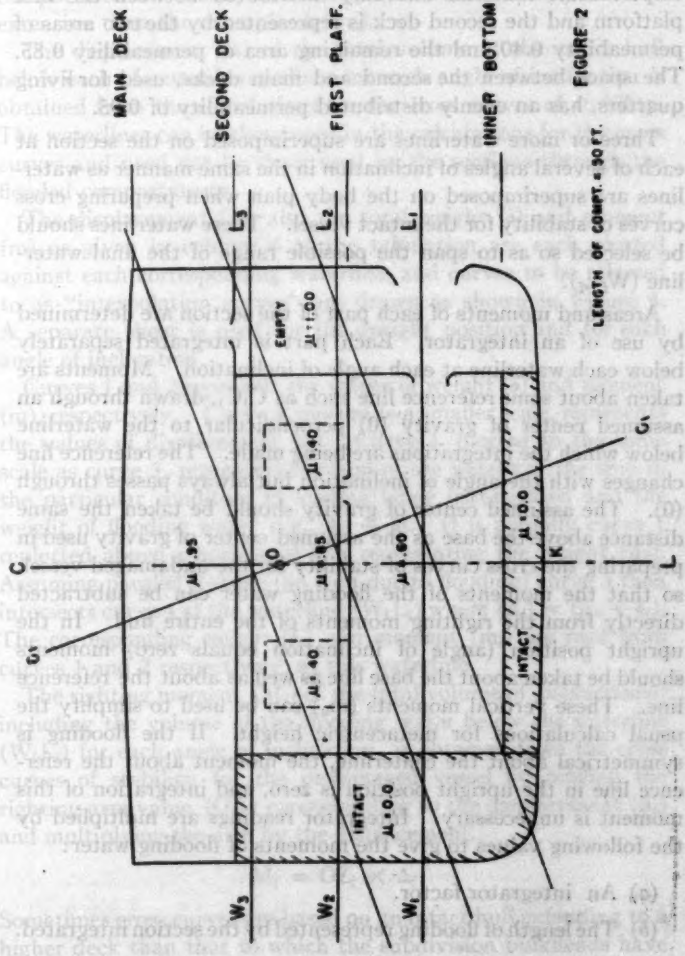
For any particular case of damage in any particular condition of loading, the final waterline after flooding ($W_f L_f$) is the waterline corresponding to a displacement (Δ_f), equal to the sum of the weight of the ship (Δ_0), and the weight (δ) of the flooding water.

Where flooding is due to a ruptured pipe line or to a small opening in the hull, and is eventually brought under control, the weight (δ) of the flooding water is easily determined. Where the damaged compartments are in free communication with the sea, the flooding continues to the level of the external waterline, which in turn changes with respect to the ship as the flooding progresses. When the vessel heels through any angle of inclination the underwater portion of the hull changes form and the external waterline may again change with respect to the ship. Consequently, the determination of the weight (δ_f) of flooding water up to the final waterline ($W_f L_f$) is quite a problem.

Several representative sections, such as the section shown in Figure 2, are taken through the flooded compartments. If the flooded compartments are near midships where there is usually little variation in transverse section, one section taken through each compartment may be sufficient. If there is appreciable variation in transverse section throughout any compartment, more sections may be necessary. Each section is subdivided into parts for which the distribution of impermeable material can be considered uniform throughout the length (l) represented by the section. Permeability (u) (or ratio of the volume floodable to the total volume) is assigned to each part of each section.

The section shown in Figure 2 represents a compartment, say 50 feet in length, with wing tanks, port and starboard, and a double bottom tank below the hold. The wing tank on one side and the double bottom tank are assumed to remain intact after damage; the rest of the compartment is assumed open to the sea.

The damaged wing tank, originally empty, floods 100% of its volume up to the final waterline, hence the permeability equals 1.00. The hold, filled with uniformly distributed cargo occupy-


MAIN DECK
SECOND DECK
FIRST PLATF.
INNER BOTTOM
FIGURE 2
LENGTH OF COMPT. = 30 FT.
L3
L2
L1
EMPTY
μ=1.00
μ=1.40
μ=.95
μ=.85
μ=.60
μ=1.0
μ=.40
INTACT
μ=0.0
INTACT
μ=0.0
C
O
K
L
G
G1
W3
W2
W1

ing 40% of its volume, floods the remaining 60% of its volume up to the final waterline or to the upper boundary of the hold, which is the first platform, hence the permeability equals 0.60. Impermeable material unevenly distributed between the first platform and the second deck is represented by the two areas of permeability 0.40 and the remaining area of permeability 0.85. The space between the second and main decks, used for living quarters, has an evenly distributed permeability of 0.95.

Three or more waterlines are superimposed on the section at each of several angles of inclination in the same manner as waterlines are superimposed on the body plan when preparing cross curves of stability for the intact vessel. These waterlines should be selected so as to span the possible range of the final waterline ($W_f L_f$).

Areas and moments of each part of the section are determined by use of an integrator. Each part is integrated separately below each waterline at each angle of inclination. Moments are taken about some reference line such as C_1C_1, drawn through an assumed center of gravity (0) perpendicular to the waterline below which the integrations are being made. The reference line changes with the angle of inclination but always passes through (0). The assumed center of gravity should be taken the same distance above the base as the assumed center of gravity used in preparing the cross curves of stability for the undamaged vessel, so that the moments of the flooding water can be subtracted directly from the righting moments of the entire hull. In the upright position (angle of inclination equals zero) moments should be taken about the base line as well as about the reference line. These vertical moments (m_v) can be used to simplify the usual calculations for metacentric height. If the flooding is symmetrical about the centerline, the moment about the reference line in the upright position is zero, and integration of this moment is unnecessary. Integrator readings are multiplied by the following values to give the moments of flooding water:

(*a*) An integrator factor.

(*b*) The length of flooding represented by the section integrated.

(*c*) The permeability of the part integrated.

(*d*) A conversion factor, including the scale of the section and the density of the flooding water.

These weights and moments are tabulated as shown in Figure 3, columns 1 to 5, and added algebraically to give the results, column 6. (Since more than one section through the flooded compartments is usually necessary, the tabulations may be more extensive than the example shown).

The displacement (Δ) of the total volume of the entire hull below each of several waterlines for each angle of inclination is obtained from the calculations for the cross curves of stability. The waterlines can be those used in the calculations for the cross curves and need not be those used on the sections through the flooded compartments.

The displacement (Δ), also the total weight (δ) and moment (m) as given in column 6 of the tabulation are each plotted against each corresponding waterline, and curves to be referred to as "interpolation curves" are drawn as shown in Figure 4. A separate sheet is used for the upright position and for each angle of inclination.

Curves 1 and 2 represent the values of weight (δ) and moment (m), respectively. Curve 3, plotted to a smaller scale, represents the values of displacement (Δ). Curve 4, plotted to the same scale as curve 3, represents the sum of the weight of the ship in the particular condition of loading being investigated and the weight of flooding water (i.e., $\Delta_0 + \delta$). It is actually curve 1 replotted above a horizontal line representing the weight (Δ_0). Assuming parallel sinkage (no trim due to flooding) curve 3 then intersects curve 4 at the waterline ($W_f L_f$) where (Δ_f) = ($\Delta_0 + \delta_f$). The corresponding weight (δ_f) and moment (m_f) are read from curves 1 and 2 respectively, at this waterline.

The righting moment (M_f) of the total volume of displacement including the volume of the flooding water below the waterline ($W_f L_f$) for each angle of inclination, is obtained from the cross curves of stability for the undamaged vessel by reading the righting arm value (OZ_f) corresponding to the displacement (Δ_f) and multiplying the arm by the displacement.

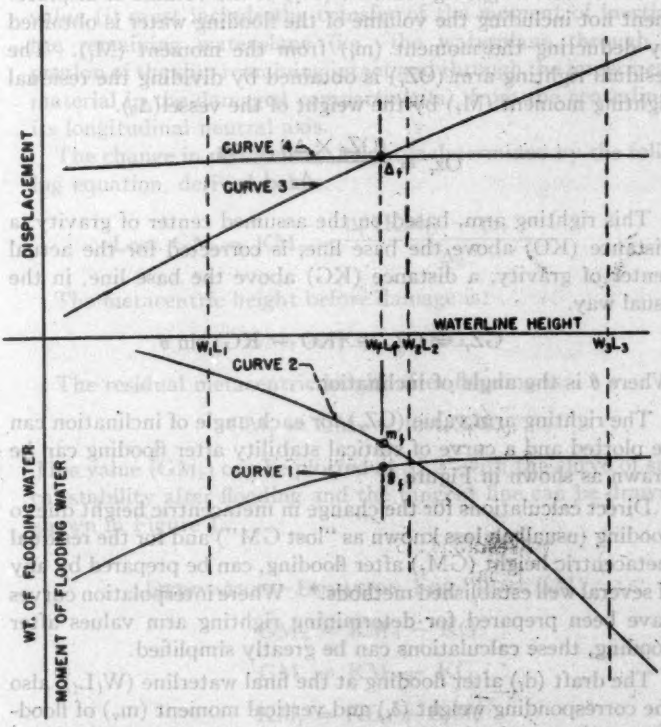
$$M_f = OZ_f \times \Delta_f$$

Sometimes cross curves are based on an intact hull extending to a higher deck than that to which the subdivision bulkheads have been made watertight. As flooding can progress throughout the space above the bulkhead deck, none of this space can be considered as part of the intact portion of the vessel after damage.

	COLUMN 1 DOUB. BOT. - FIRST PL. $\mu = .60$		COLUMN 2 FIRST PL. - SEC. DK. $\mu = .85$		COLUMN 3 BLOCKS, P/S $\mu = .40$		COLUMN 4 WING TANK $\mu = 1.00$		COLUMN 5 SEC. DK. - MAIN DK. $\mu = .95$		COLUMN 6 TOTAL	
	WT.	MOM.	WT.	MOM.	WT.	MOM.	WT.	MOM.	WT.	MOM.	WT.	MOM.
W_1L_1 0°		VERT. TRANS.		VERT. TRANS.		VERT. TRANS.		VERT. TRANS.		VERT. TRANS.		VERT. TRANS.
10°		TRANS.										
20°		TRANS.										
30°		TRANS.										
40°		TRANS.										
50°		TRANS.										
W_2L_2 0°		VERT. TRANS.		VERT. TRANS.		VERT. TRANS.		VERT. TRANS.		VERT. TRANS.		VERT. TRANS.
10°												
20°												
30°												
40°												
50°												
W_3L_3 0°		VERT. TRANS.		VERT. TRANS.		VERT. TRANS.		VERT. TRANS.		VERT. TRANS.		VERT. TRANS.
10°												
20°												
30°												
40°												
50°												

* (EXCL. OF BLOCKS)

FIGURE 3



FOR:

Δ_1 = WEIGHT OF THE SHIP IN THE SPECIFIC CONDITION OF LOADING BEING INVESTIGATED

KO = HEIGHT OF THE ASSUMED CENTER OF GRAVITY ABOVE THE BASE LINE

θ_1 = ANGLE OF INCLINATION

FIGURE 4

The entire space must be included in the calculations for the weight and moment of the flooding water, or else the cross curves for the undamaged vessel must be revised to include only the intact hull up to the bulkhead deck.

The residual righting moment (M_r) of the volume of displacement not including the volume of the flooding water is obtained by deducting the moment (m_f) from the moment (M_f). The residual righting arm (OZ_r) is obtained by dividing the residual righting moment (M_r) by the weight of the vessel(Δ_0).

$$Oz_r = \frac{(OZ_f \times \Delta_f) - m_f}{\Delta_0}$$

This righting arm, based on the assumed center of gravity, a distance (KO) above the base line, is corrected for the actual center of gravity, a distance (KG) above the base line, in the usual way.

$$GZ_r = OZ_r + (KO - KG) \sin \theta$$

Where θ is the angle of inclination.

The righting arm value (GZ_r) for each angle of inclination can be plotted and a curve of statical stability after flooding can be drawn as shown in Figure 1.

Direct calculations for the change in metacentric height due to flooding (usually a loss known as "lost GM") and for the residual metacentric height (GM_r) after flooding, can be prepared by any of several well established methods.* Where interpolation curves have been prepared for determining righting arm values after flooding, these calculations can be greatly simplified.

The draft (d_f) after flooding at the final waterline ($W_f L_f$), also the corresponding weight (δ_f) and vertical moment (m_v) of flooding water are obtained from the interpolation curves for the upright position. The height (KM_f) of the transverse metacenter above the base for the total volume of displacement below the final waterline (including the volume of the flooding water) is read at the draft (d_f) after flooding from the curves of form prepared for the intact undamaged vessel. The height (KM_0) of the transverse metacenter above the base for the vessel before damage, is read at the draft (d_0) before damage from the same curves of form.

*Ibid pages 185-186.

The loss in moment of inertia of the waterplane, designated as (i), is the transverse moment of inertia of the surface of the flooding water at the level of the final waterline ($W_f L_f$). Where this surface is symmetrical, the moment of inertia (i) is taken about its centerline. Where this surface is not symmetrical, the value (i) must include the transfer of the moment of inertia of the remaining waterplane (i.e., the waterplane through the portion of the ship remaining intact and through the impermeable material in the damaged compartments), from the centerline to its longitudinal neutral axis.

The change in metacentric height is determined by the following equation, derived below.

$$\text{Lost GM} = KM_0 - \frac{\Delta_f (KM_f) - m_v - i}{\Delta_0}$$

The metacentric height before damage is:

$$GM_0 = KM_0 - KG$$

The residual metacentric height after flooding is:

$$GM_r = GM_0 - \text{lost GM}$$

This value (GM_r) can be plotted at 57.3° with the curve of statical stability after flooding and the tangent line can be drawn as shown in Figure 1.

DERIVATION—EQUATION FOR "LOST GM"

$$GM_0 = KM_0 - KG \tag{1}$$

$$GM_r = KM_r - KG \tag{2}$$

$$KM_r = KB_r + B_rM_r \tag{3}$$

$$KM_f = KB_f + B_fM \tag{4}$$

$$B_rM_r = \frac{I_f - i}{\Delta_0} \tag{5}$$

$$B_fM_f = \frac{I_f}{\Delta_f} \tag{6}$$

Subtracting equation (2) from equation (1),

$$\text{Lost GM} = GM_0 - GM_r = KM_0 - KM_r \tag{7}$$

Combining the moments of buoyancy of the intact portion and the flooded portion of the damaged vessel,

$$\Delta_f (KB_f) = \Delta_0 (KB_r) + m_v \tag{8}$$

transposing,

$$KB_r = \frac{\Delta_f (KB_f) - m_v}{\Delta_0} \tag{9}$$

transposing equation (6),

$$I_f = \Delta_f (B_fM_f) \tag{10}$$

substituting equation (10) in equation (5),

$$B_rM_r = \frac{\Delta_f (B_fM_f) - i}{\Delta_0} \tag{11}$$

substituting equation (9) and (11) in equation (3),

$$KM_r = \frac{\Delta_f (KB_f) - m_v}{\Delta_0} + \frac{\Delta_f (B_fM_f) - i}{\Delta_0} \tag{12}$$

$$KM_r = \frac{\Delta_f (KB_f + B_fM_f) - m_v - i}{\Delta_0} \tag{13}$$

substituting equation (4) in equation (13),

$$KM_r = \frac{\Delta_f (KM_f) - m_v - i}{\Delta_0} \tag{14}$$

substituting equation (14) in equation (7),

$$\text{Lost GM} = KM_0 - \frac{\Delta_f (KM_f) - m_v - i}{\Delta_0}$$

General Studies

In considering all of the different cases of flooding that might result from the extent of damage a vessel could sustain without exceeding the floodable length, and all of the probable conditions of loading in which the vessel might operate, the many studies already made serve as an indication of certain cases and conditions that are sure to be less critical than others. In this respect everything else being equal, the following is generally true:

(*a*) The flooding of wide compartments causes a greater loss in stability than the flooding of narrower ones.

(*b*) The flooding of a compartment with more impermeable material near the bottom than near the top, or near the centerline than near the shell, causes a greater loss in stability than the flooding of the same compartment when the impermeable material is more uniformly distributed.

(*c*) Flooding occurring in the lightest condition of loading in which a vessel operates, or in the full load condition (approaching the limiting drafts determined from a study of the subdivision plan), results in less stability after damage than flooding occurring in any intermediate condition.

Stability characteristics for all cases and conditions not obviously less critical than others must be calculated. A separate study of each case and condition would involve a discouraging amount of work, but this work can be greatly reduced by studying all cases together and applying many of the same calculations to several different cases; also by studying only a few arbitrary conditions of loading (spanning the operating range of the vessel) and interpolating the results for the actual conditions. In this way all studies can be made and completed before any actual conditions are determined.

Drafts before damage for the arbitrary conditions of loading are selected. At least three drafts should be used. Representative sections through each compartment considered flooded in any case of damage are prepared. Waterlines, estimated to approximate those on which the vessel may float after flooding from the drafts selected, are superimposed on each section at each of several angles of inclination. The procedure is then generally similar to that previously described.

Each part of each section is integrated below each waterline. A separate tabulation of weights and moments of flooding water is prepared for each case of flooding. The permeabilities assigned to some columns of the tabulations are different for the different waterlines consistent with the conditions of loading before damage corresponding to the waterlines.

The weights (δ) and moments (m) of flooding water, determined and recorded in the last column of each tabulation, are plotted, and interpolation curves for each angle of inclination are prepared. Curves of the sums of the weight of the vessel and the weight of flooding water ($\Delta_0 + \delta$) are prepared for the weight

(Δ_0) corresponding to each draft before damage. These curves intersect the respective curves of total displaced volume (Δ) at waterlines after flooding corresponding to each draft before damage. The total volumes of displacement (Δ_f) and moments (m_f) of flooding water up to these waterlines are then determined.

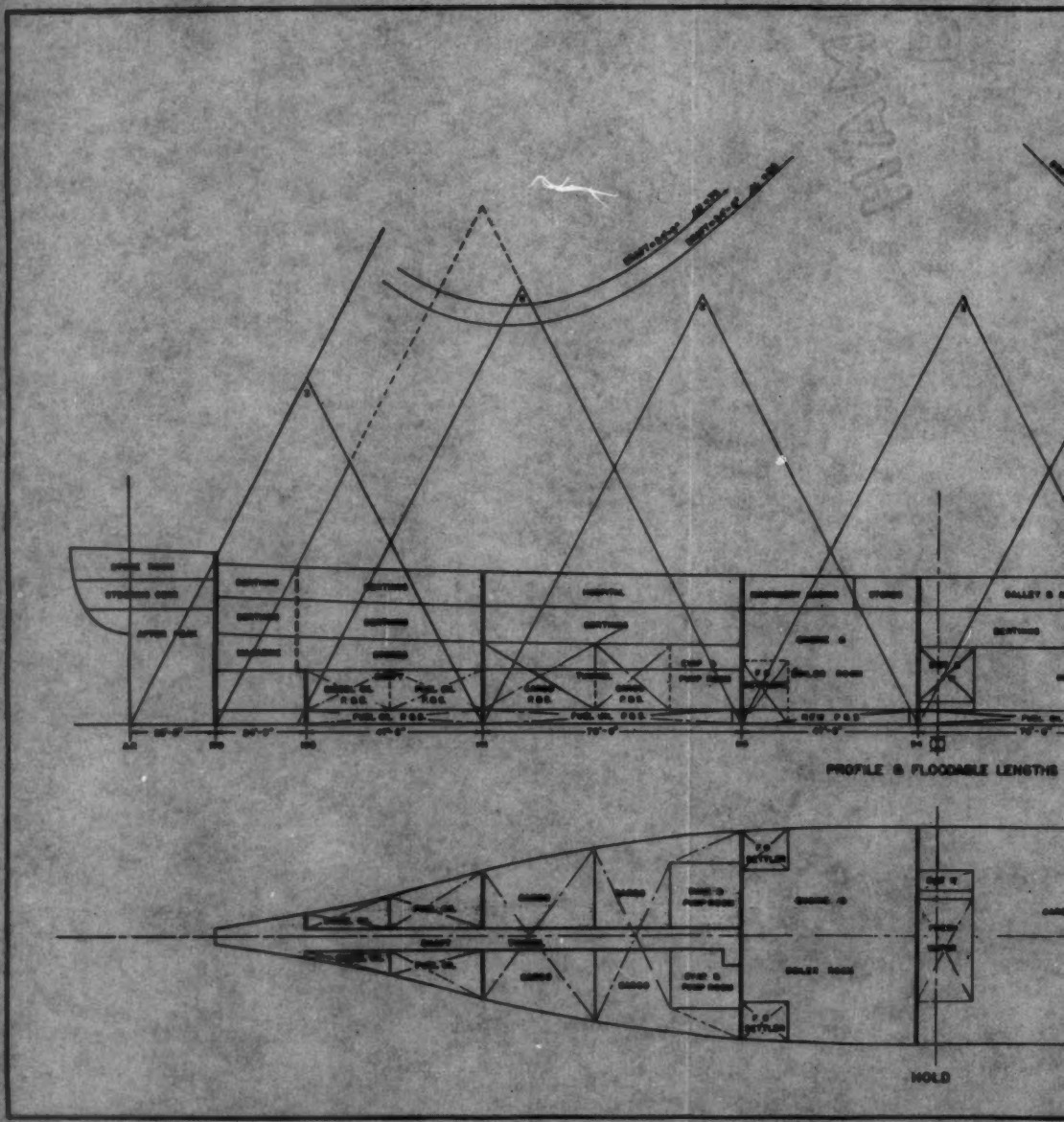
Residual righting arms are calculated and plotted as curves of statical stability after flooding. "Lost GM" values and residual metacentric heights are calculated and tangent lines are drawn. For the purpose of general studies, two curves of residual righting arms are usually prepared for each draft before damage and each case of flooding. These curves are based on arbitrary centers of gravity, one foot apart, selected so that the stability characteristics indicated by one of these curves will be about the minimum acceptable, and those indicated by the other will be somewhat less than acceptable. Actual conditions for which the centers of gravity are either lower or higher than the arbitrary centers are then obviously satisfactory or unsatisfactory respectively.

Righting arms before damage are determined and plotted with the corresponding curves of residual righting arms but only one curve, based on the lower of the two arbitrary centers, is drawn for each draft and case of flooding. The loss in stability is then clearly indicated by the difference between the curve of intact righting arms and the higher of the two curves of residual righting arms.

A comparison of the curves for the different cases of flooding will show which case is most critical. It will sometimes be found that one case is most critical at light drafts or small angles of inclination and another case is most critical at deeper drafts or large angles of inclination. Only the curves for the most critical cases are of interest once they are determined.

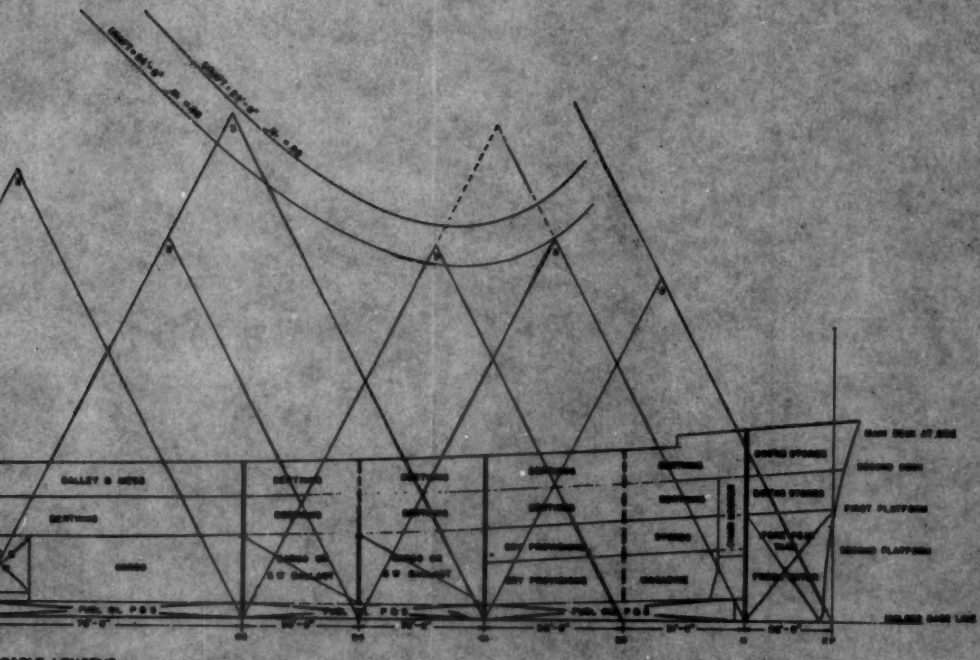
A comprehensive indication of the stability characteristics for any actual condition of loading can be readily obtained by inspection of the curves for the arbitrary conditions, but specific curves of statical stability are usually desired. Residual righting arms for the arbitrary conditions are plotted against the corresponding drafts before damage and cross curves after flooding are drawn. Values are then read back at the actual draft (d_0) and corrected for the actual height (KG) of the center of gravity above the base.

Plate II shows the complete results of the general studies of

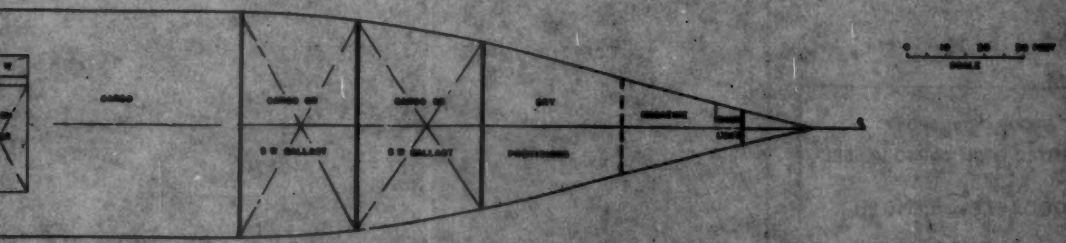


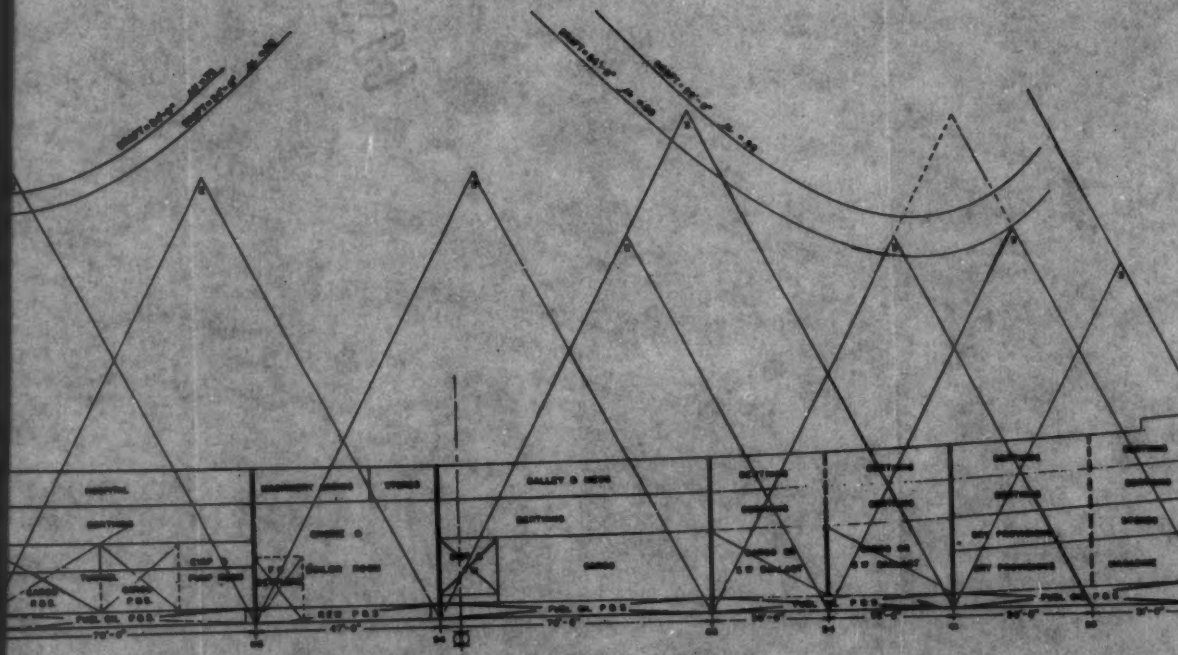
PROFILE & FLOODABLE LENGTHS

HOLD

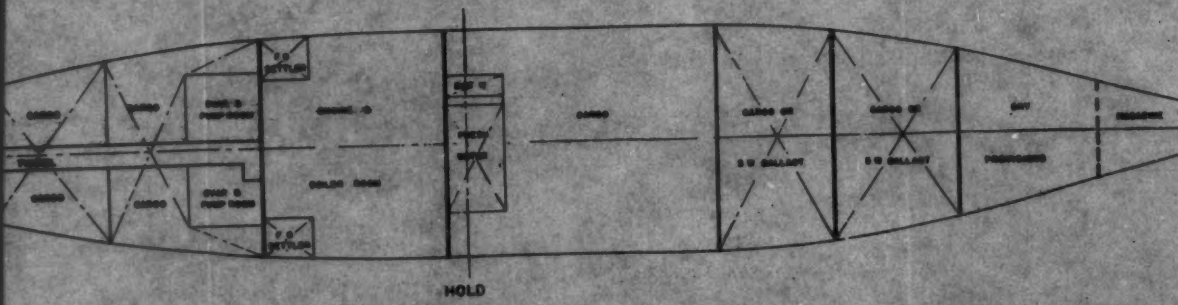


CABLE LENGTHS



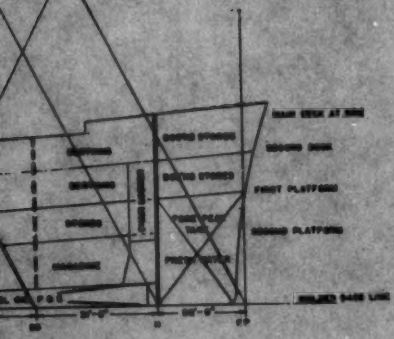


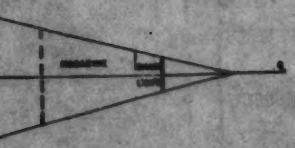
PROFILE & FLOODABLE LENGTHS



HOLD

PRINCIPAL CHARACTERISTICS

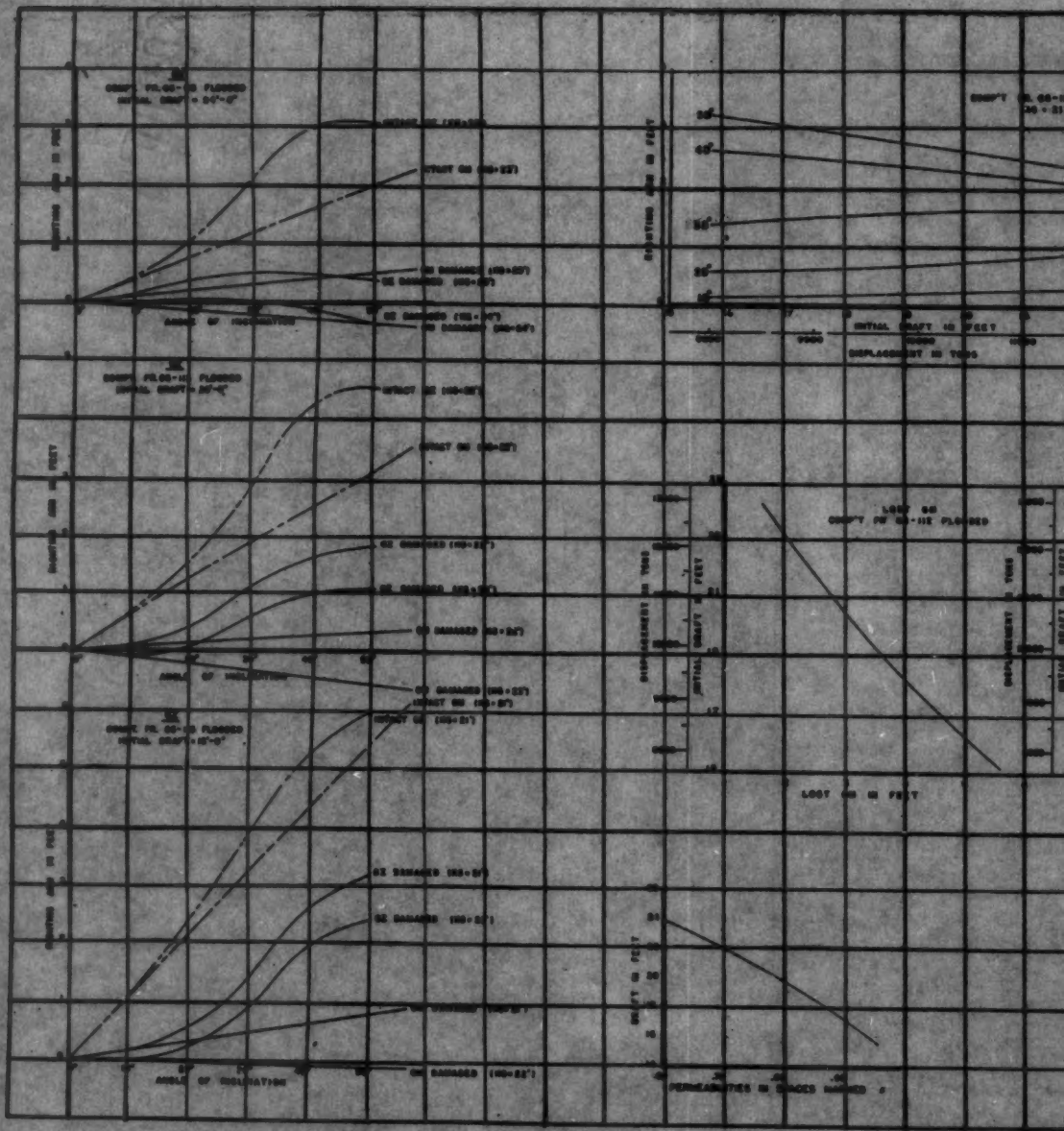


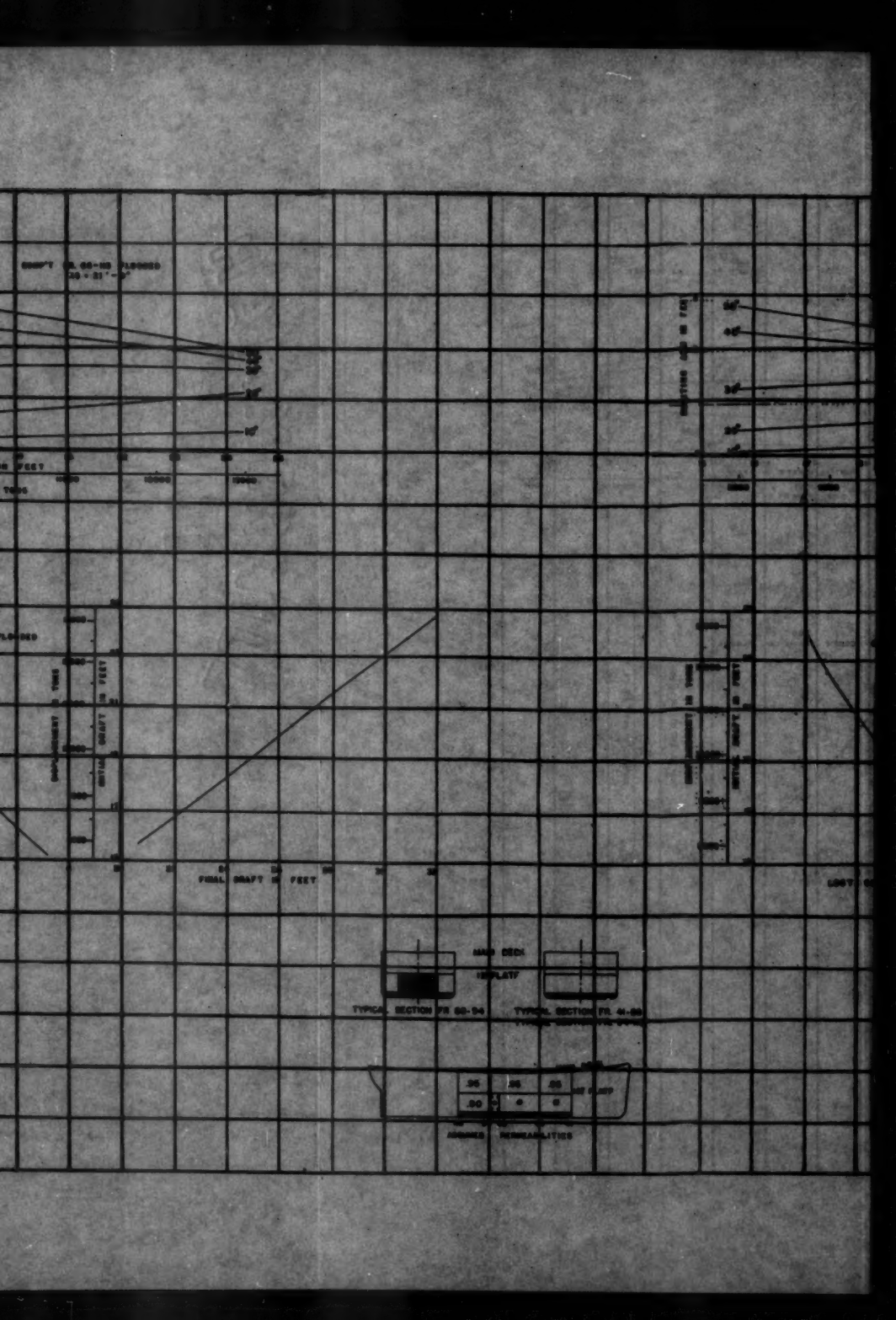


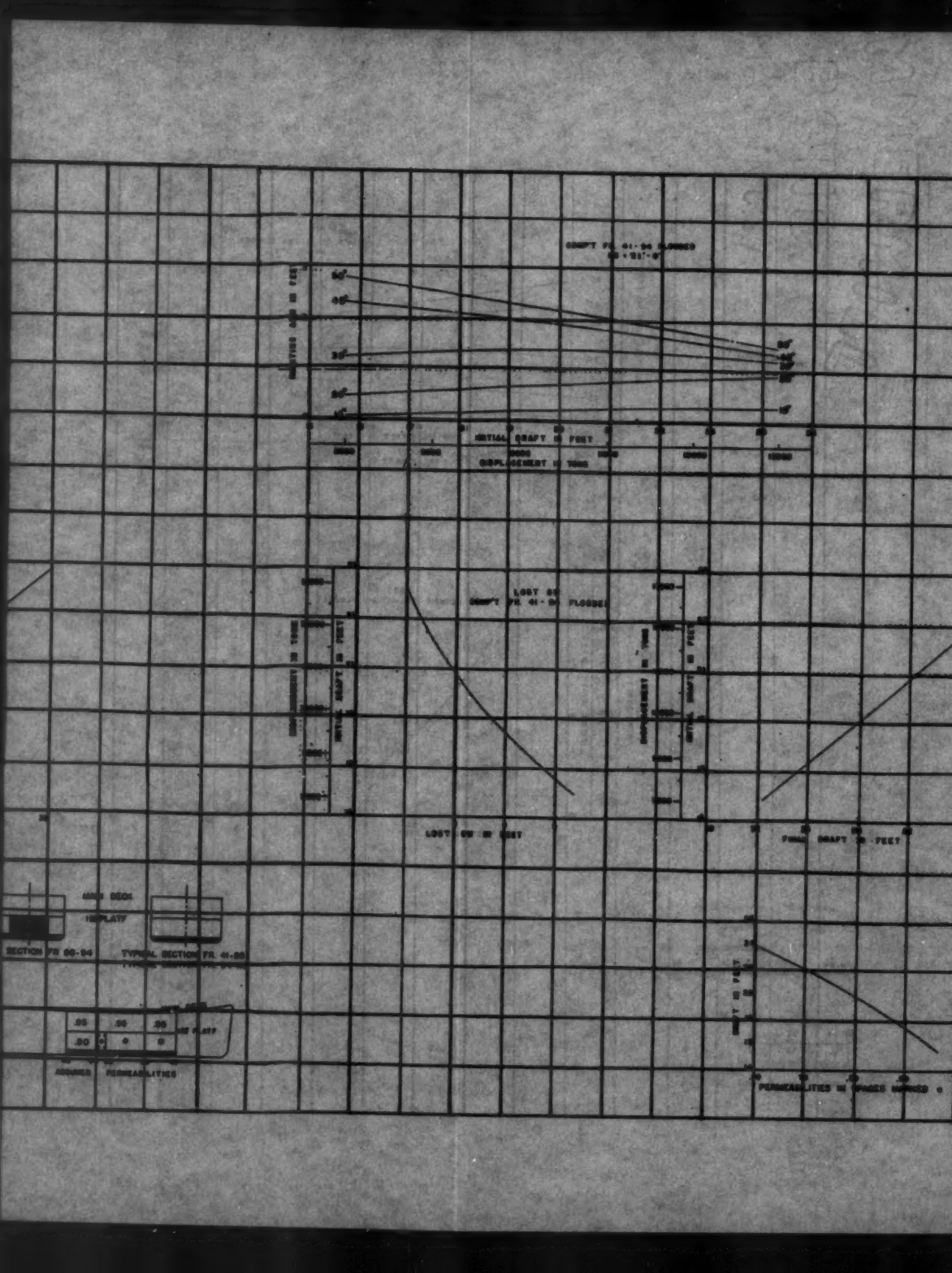
TROOP TRANSPORT
CONVERTED FROM CARGO VESSEL

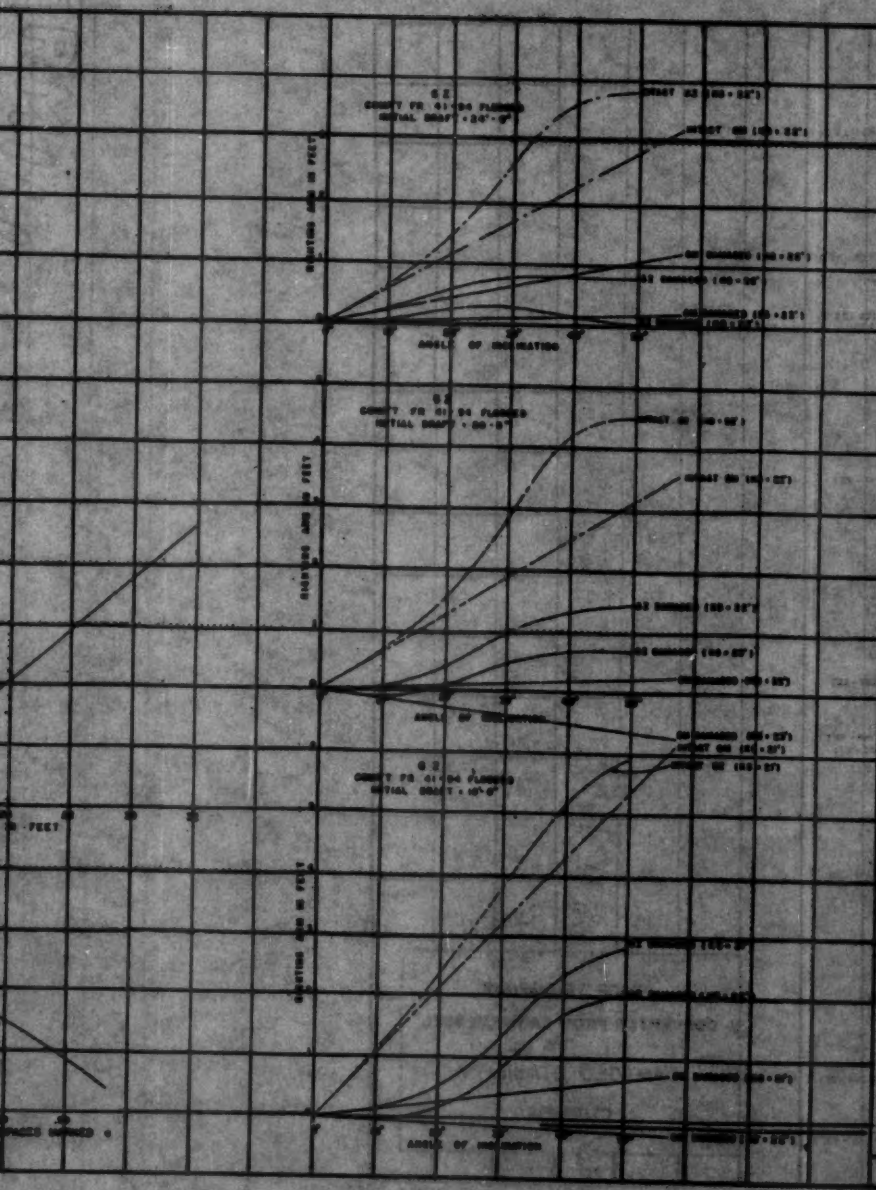
FLOODABLE LENGTH CURVES

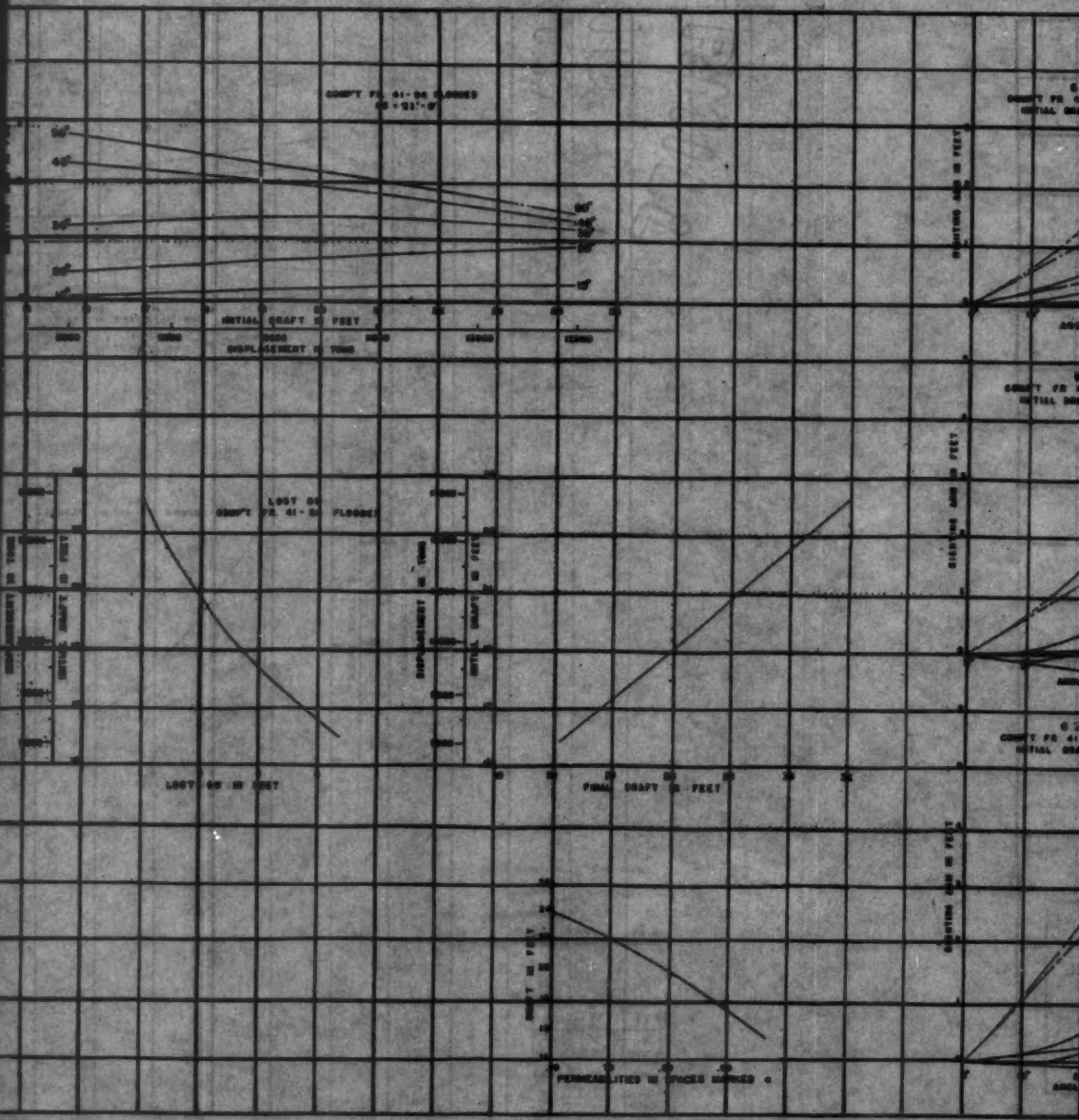
PLATE I

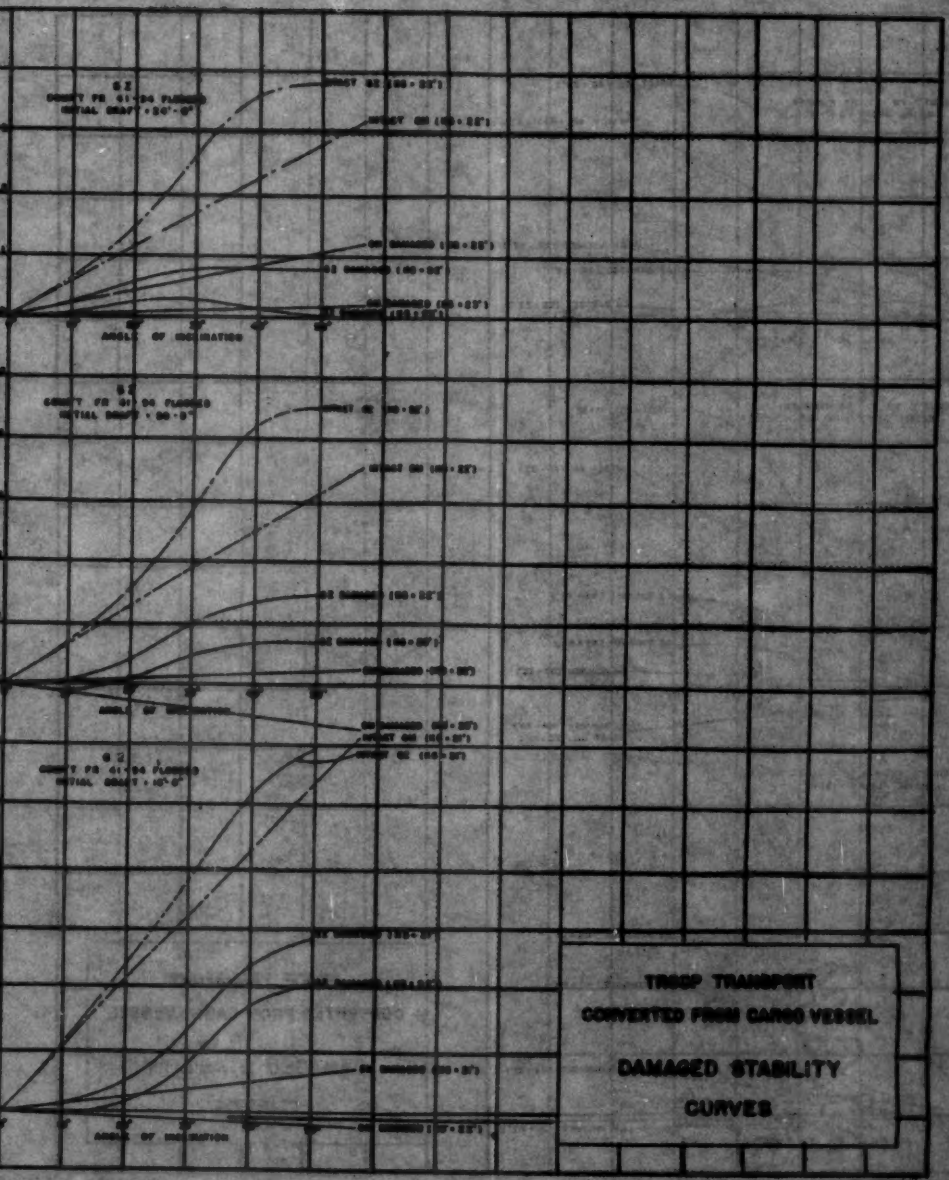



FINAL DRAFT IN FEET
MAIN DECK
TYPICAL SECTION FR 60-94
TYPICAL SECTION FR 41-59
.95
.85
.85
.90
.0
.0
ASSUMED PERMEABILITIES









TROOP TRANSPORT
CONVERTED FROM CARGO VESSEL

DAMAGED STABILITY
CURVES

PLATE II

two different cases of flooding for the troop transport shown on Plate I, presented in convenient form for ready application to any actual condition of loading. The flooding is shown by sketches of the compartments involved. Permeabilities assumed in these compartments are indicated on the sketches or plotted against the drafts before damage to which they correspond. Righting arms before and after flooding are plotted as curves of statical stability for each arbitrary draft upon which the studies were based. Righting arms after flooding are also plotted as cross curves. A curve of "lost GM" is included, mainly for comparison with older studies. Drafts after damage are plotted against corresponding drafts before damage to show the residual freeboard and to facilitate determining the angle of inclination at which the bulkhead deck becomes submerged, or progressive flooding begins. A comparison of the results reveals that the flooding from frame 41 to 94 is more critical than the flooding from frame 66 to 113, but this could not have been determined unless studies for both cases had been made. Ordinarily the less critical case would not be included on the plan.

Cross Curves of Stability After Flooding

The procedure for a general study can be modified in certain respects so that the interpolation curves can be eliminated and cross curves of stability after flooding can be prepared directly. There is no appreciable difference in the total amount of work involved. The choice is mainly a matter of personal preference.

In the modified procedure* sections taken through the damaged compartments are prepared as before. At least four waterlines are superimposed on each section. The weights and moments of flooding water below each waterline are determined, tabulated, and summed up. Each value of the weight (δ), given by the last column of the tabulation, is subtracted from the corresponding total volume of displacement (Δ_f) to give the weight (Δ_0) of the vessel, i.e., the displacement before damage that would result in flooding to that waterline. The value (Δ_0) is usually different for each waterline and each angle of inclination. The residual righting arm (OZ_r) is calculated for each value of (Δ_0) and plotted against the corresponding draft (d_0) before damage. Cross curves of stability after flooding are then drawn.

*Described in Navy Department, Bureau of Ships, Technical Memorandum No. 1, "Subdivision of Ships and Stability after Extensive Flooding," Restricted, Washington, D. C., now being prepared.

Intact Spaces in Damaged Compartments.

Where it is desired to determine stability characteristics for the worst possible case of damage, it is logical to assume that bulkheads on the opposite side of the centerline from the damage, and that platforms or flats low in the damaged compartments, remain intact. Wing tanks and deep tanks on the opposite side of the centerline from the damage are accordingly considered to remain intact while those on the damaged side are considered to be open to the sea. Shallow centerline tanks are usually considered to remain intact. The intact spaces formed are blocked out on the sections representative of the flooded compartments and are omitted from the integrations.

Tanks that are slack or empty, or that are full of liquids less dense than the flooding water, and that do not extend above the final waterline after flooding, will flood unsymmetrically if opened to the sea by the damage. The weight and the moment of this flooding water must be included in the integrations.* This necessitates revising one or more columns of the tabulation, and preparing an additional set of interpolation curves or cross curves for each different condition of the tanks.

Ordinarily tanks located unsymmetrically about the centerline are kept as full as practicable in order to minimize the unsymmetrical flooding that might occur. In the general studies, only those tanks in the damaged compartments that might be empty at any one time, and that if flooded unsymmetrically would cause the greatest heeling moment (usually one pair of tanks of each type of liquid), are considered to be empty. All other tanks in the damaged compartments are considered to be full. The weight estimate for the condition of loading before damage (d_0, KG) must be based on the same condition of the tanks as the calculations for the weights and moments of the flooding water.

In the upright position the flooding of unsymmetrical spaces causes a heeling moment towards the damaged side. The vessel heels to some angle where the heeling moment is reduced to zero. The residual righting moments are accordingly less than if the tanks had remained intact.

Tanks extending well above the external waterline and containing liquids above that level will allow some liquids to run off if opened to the sea by the damage. The weight and moment of this run-off must be included in the integrations.*

*Ibid.

In the upright position, run-off causes a heeling moment away from the damaged side. The vessel heels to some angle where the heeling moment is reduced to zero. If the vessel continues to roll in the direction of the heel, the run-off will increase with the angle of inclination, and the residual righting moments will be accordingly less than if the tanks had remained intact.

The surface of the flooding water used in calculating the "lost GM" and the residual metacentric height (GM_r), extends through the damaged tanks and through any spaces opened to the sea, but does not extend through any intact tanks or spaces.

In drawing the GM tangent line to the righting arm curve where the righting arm in the upright position is not zero, the residual metacentric height (GM_r) plotted at 57.3° is laid off above the righting arm value at 0° rather than above the horizontal axis of the plot. The value laid off above the horizontal axis is ($GM_r + GZ_r$ at 0°) or (GM_r − heeling arm).

Effect of Trim.

Factors, such as change in trim, that must be neglected in studies of stability after flooding if the calculations are not to become too involved to be practical, should be clearly borne in mind if the principles involved are to be understood and if special cases where the effects of these factors may be significant are to be recognized.

Damage near the bow or stern of a vessel will cause an appreciable change in trim which cannot be justifiably neglected, but the compartments near the bow or stern are comparatively narrow and their flooding need seldom be considered with the critical cases.

The model tests,* conducted in conjunction with the development of the methods described, were based on both normal trim and parallel sinkage. A comparison of the results showed very little effect of trim on righting arms in any case. However, the compartments flooded did not extend very far into the ends of the model and no great changes in trim occurred. A special test was conducted with large trims of four and five percent of the length of the model imposed, and the results of this test are shown in Figures 5 and 6. Righting arms and mean drafts based on parallel sinkage are indicated by the solid lines. Righting arms for trim by the bow, and drafts forward, are indicated

*Ibid.

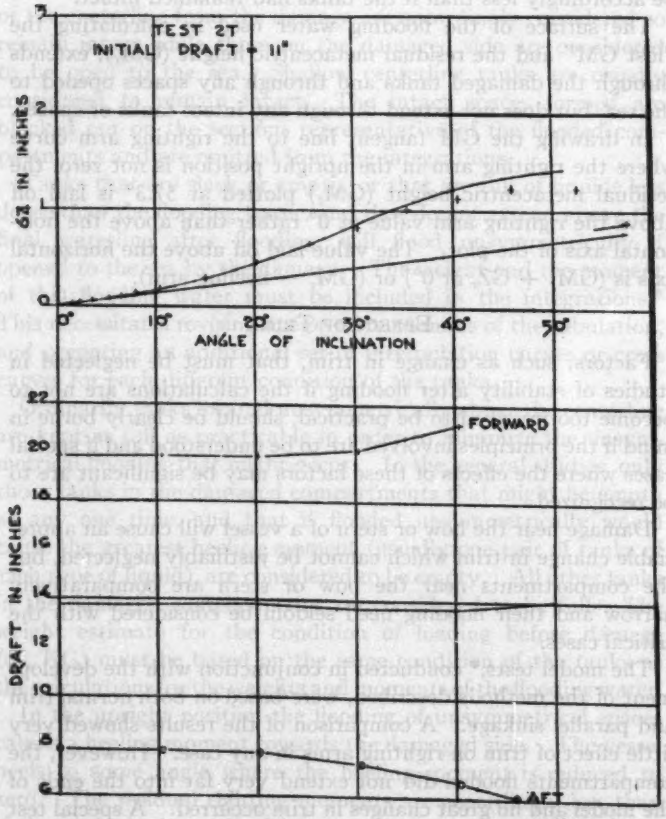

TEST 2T
INITIAL DRAFT = 11"
GM IN INCHES
3
2
1
0
0°
10°
20°
30°
40°
50°
ANGLE OF INCLINATION
FORWARD
AFT
DRAFT IN INCHES
22
20
18
16
14
12
10
8
6

FIGURE 8

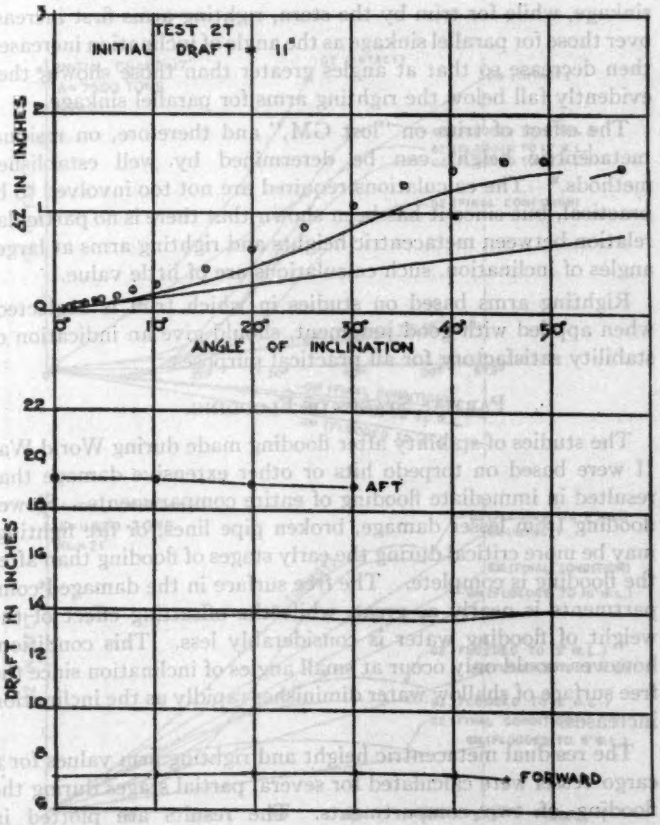

TEST 2T
INITIAL DRAFT = 11'
GZ IN INCHES
ANGLE OF INCLINATION
AFT
FORWARD
DRAFT IN INCHES

FIGURE 6.

by (+). Righting arms for trim by the stern, and drafts aft, are indicated by (•). For trim either by the bow or by the stern, the metacentric height (or initial stability) is shown to be greater than that for parallel sinkage. For trim by the bow, righting arms at large angles of inclination are less than those for parallel sinkage, while for trim by the stern, righting arms first increase over those for parallel sinkage as the angle of inclination increases, then decrease so that at angles greater than those shown, they evidently fall below the righting arms for parallel sinkage.

The effect of trim on "lost GM," and therefore, on residual metacentric height can be determined by well established methods.* The calculations required are not too involved to be practical, but since it has been shown that there is no particular relation between metacentric heights and righting arms at larger angles of inclination, such calculations are of little value.

Righting arms based on studies in which trim is neglected, when applied with good judgment, should give an indication of stability satisfactory for all practical purposes.

Partial Stages of Flooding.

The studies of stability after flooding made during World War II were based on torpedo hits or other extensive damage that resulted in immediate flooding of entire compartments. Slower flooding from lesser damage, broken pipe lines, or fire fighting, may be more critical during the early stages of flooding than after the flooding is complete. The free surface in the damaged compartments is nearly as great, while the offsetting effect of the weight of flooding water is considerably less. This condition, however, could only occur at small angles of inclination since the free surface of shallow water diminishes rapidly as the inclination increases.

The residual metacentric height and righting arm values for a cargo vessel were calculated for several partial stages during the flooding of two compartments. The results are plotted in Figure 7. This figure shows that losses in metacentric height and righting arm values at small angles of inclination are greater in the early stages of flooding than as the flooding progresses. At larger angles, the righting arm values decrease continually as the flooding progresses, and the worst condition occurs when

*Rossell, Henry E. and Chapman, Lawrence B. "Principles of Naval Architecture," Volume I, pages 183, 184, 187-189, The Society of Naval Architects and Marine Engineers, 1939.

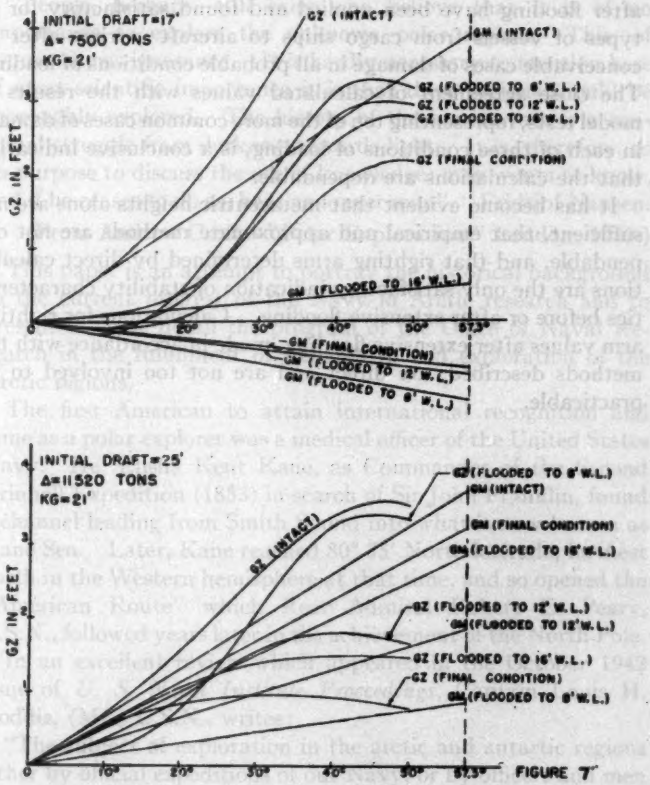

INITIAL DRAFT=17'
Δ=7500 TONS
KG=21'
GZ IN FEET
GZ (INTACT)
GM (INTACT)
GZ (FLOODED TO 8' W.L.)
GZ (FLOODED TO 12' W.L.)
GZ (FLOODED TO 16' W.L.)
GZ (FINAL CONDITION)
GM (FLOODED TO 16' W.L.)
GM (FINAL CONDITION)
GM (FLOODED TO 12' W.L.)
GM (FLOODED TO 8' W.L.)
10°
20°
30°
40°
50°
57.3°
INITIAL DRAFT=25'
Δ=11520 TONS
KG=21'
GZ IN FEET
GZ (FLOODED TO 8' W.L.)
GM (INTACT)
GZ (INTACT)
GM (FINAL CONDITION)
GM (FLOODED TO 16' W.L.)
GZ (FLOODED TO 12' W.L.)
GM (FLOODED TO 12' W.L.)
GZ (FLOODED TO 16' W.L.)
GZ (FINAL CONDITION)
GM (FLOODED TO 8' W.L.)
10°
20°
30°
40°
50°
57.3°

FIGURE 7

flooding is complete. It is logical to assume that the same is true in nearly all vessels, and consequently partial stages of flooding need be considered only for the effect on initial stability.

In Conclusion.

The methods described for studying subdivision and stability after flooding have been applied and found satisfactory for all types of vessels from cargo ships to aircraft carriers after all conceivable cases of damage in all probable conditions of loading. The close agreement of calculated values with the results of model tests, representing ten of the more common cases of damage in each of three conditions of loading, is a conclusive indication that the calculations are dependable.

It has become evident that metacentric heights alone are not sufficient, that empirical and approximate methods are not dependable, and that righting arms determined by direct calculations are the only satisfactory indication of stability characteristics before or after extensive flooding. Calculations for righting arm values after extensive flooding, made in accordance with the methods described, are direct and are not too involved to be practicable.

THE NAVY EXPLORES ITS NORTHERN FRONTIERS.

By Dr. M. C. Shelesnyak.*

"People, perhaps, still exist who believe that it is of no importance to explore the unknown polar regions. This, of course, shows ignorance. It is hardly necessary to mention here of what scientific importance it is that these regions should be thoroughly explored. The history of the human race is a continual struggle from darkness towards light. It is, therefore, to no purpose to discuss the use of knowledge; man wants to know, and if he ceases to do so, he is no longer man". Fridtjof Nansen.

(From Andrew Croft—"Polar Exploration," London, 1939.)

This paper is an attempt to portray the historical background to the current interest of the Navy in Arctic research and to present in some detail the program of the Office of Naval Research in the fulfillment of research in and exploration of the Arctic regions.

The first American to attain international recognition and fame as a polar explorer was a medical officer of the United States Navy. He, Elisha Kent Kane, as Commander of the Second Grinnell Expedition (1853) in search of Sir John Franklin, found a channel leading from Smith Sound into what is now known as Kane Sea. Later, Kane reached 80° 35′ North latitude, farthest north in the Western hemisphere at that time, and so opened the "American Route" which Rear Admiral Robert E. Peary, U.S.N., followed years later in the achievement of the North Pole.

In an excellent review which appeared in the October 1942 issue of *U. S. Naval Institute Proceedings*, Captain Louis H. Roddis, (MC) U.S.N., writes:

"The subject of exploration in the arctic and antartic regions either by official expeditions of our Navy, or by officers and men of the Navy as leaders or members of private expeditions is one that has been to a large degree neglected by the naval historian. Yet, from the standpoint of the extent and importance of geographic and scientific discovery, it constitutes one of the most prominent chapters in the history of the exploration of the globe.

*Head, Environmental Biology Branch, Medical Sciences Division, Office of Naval Research, Washington, D. C.

Among the principal achievements in the field of polar exploration in which officers and men of the Navy have participated may be included the virtual discovery of the Antartic Continent; the discovery of the North Pole; and air flights over both the North and South Pole. Many important additions to the maps of the world have been made, and scientific knowledge of vast importance has been accumulated by the enterprise of the United States Navy in this field. All of this work has been done within a little more than the span of a century, for prior to 1836, the Navy had not taken any part in polar exploration."

The Northward course of exploration of the United States Navy was led by Lieutenant E. J. DeHaven, U.S.N. in 1851 (The First Grinnell Expedition). The brilliant work of Passed Assistant Surgeon Kane, in addition to reaching a "farthest north", included charting Grinnell Lane, and of singular interest, the establishment of a scientific station when the vessel under his command, *The Advance*, was wintered. Biological data on flora and fauna, keen medical observations, and extensive meteorologic, magnetic and glaciological information were collected. The material was reported by Kane, in such literary excellence of style, as has rarely been approached by explorers.

Charles Francis Hall was provided a Naval tug of 400 tons by the U. S. Navy. This he strengthened, sheathed and renamed *Polaris*. On July 3, 1871, she sailed northward under instructions of the Secretary of the Navy to attain the position of the North Pole. But Hall died near "Thank God Harbor". The *Polaris* was wrecked and part of the crew drifted 1300 miles on pack ice, to be rescued by the U.S.S. *Tigress* (under command of James A. Greer, U.S.N.) off Labrador. This drift revealed a great deal regarding the nature of polar ice behavior and formed its basis for Nansen's and Amundsen's Ship-drifts; and recently (1937) the Soviet Papanin Expedition which drifted on the ice from the Pole to North Atlantic waters.

The next in line of exploratory missions was the famous and tragic voyage of the *Jeannette*, under Commander George W. DeLong, U.S.N. DeLong established the fact that Wrangell Land was an island, and not the Southern extension of a polar land area as was conjectured. The tragic deaths of DeLong, Dr. James Ambler, his medical officer, and their party of ten, can be somewhat mitigated by the valuable scientific work which they accomplished. Melville, Chief Engineer, wrote, "*In the Lena*

Delta" which Doctor Roddis credits as one of the best books on the whole *Jeannette* Expedition. The open sea north of Bering Straits was strongly suggested, if not revealed, by DeLong.

Under the command of W. S. Schley, two U. S. Navy relief ships, *Thetis* and *Bear* were sent into the Arctic waters to seek the Greely Expedition, and succeeded in reaching Lieutenant A. W. Greely of the U. S. Army, at Bedford Pim Island on June 22, 1844. Although principally a rescue mission, the operation permitted an extensive amount of exploration.

Robert E. Peary, Rear Admiral, Civil Engineer Corps, of the U. S. Navy, is perhaps the most outstanding of the Navy explorers. Although his singular goal, the attainment of 90° North Latitude, which he accomplished on April 6, 1909, after almost a lifetime of effort, is most notable, Peary made many less heralded scientific contributions to Arctic science.

Peary's contribution to the fund of knowledge of the Arctic was concentrated on the northern Greenland area where nearly all of his important Arctic journeys were made or which he used as a base. Peary discovered the Cape York, Greenland, collection of meteorites. He emphasized the value of meteorological observations in Greenland as a basis for weather forecasting. He was one of the discoverers of the Greenland ice cap wind system, and he developed a method of pack ice sled travel.

Just less than two decades after Peary's achievement of the North Pole, another officer of the U. S. Navy, Rear Admiral Richard Evelyn Byrd, (Retired) furnished with planes and personnel by the Navy Department, made a successful flight from northern Spitsbergen and on May 9, 1926 at 2102 GST flew over the North Pole. With him was Floyd Bennett who was loaned to him by the Navy as pilot mechanic. Three years later Byrd made a successful flight over the South Pole.

While it is true that Byrd's expeditions were not dierctly under the Navy Department, he was an officer in the U. S. Navy and the Navy Department has always supported, in part, his expeditions, and thus these Arctic and Antartic expeditions must truly be included in the Navy's historical role in Polar exploration.

Add to this brief list the explorations of Lieutenant Commander Fitzhugh Greene of the Crocker Land expedition and Commander Donald B. MacMillan, U.S.N.R., who led the Crocker Land expedition and later made more than twenty Arctic journeys into Iceland, Greenland, Baffin Land and Elles-

mere Land; MacMillan's expeditions were private enterprises, but he was a Reservist in the U. S. Navy, and worked closely with the Navy. Also worthy of note are the explorations of Lieutenant Commander Isaac Schlossback; Lieutenant Commander, (now Rear Admiral) Richard H. Cruzen; Captain L. L. Adamkiewicz, (MC), U.S.N.; and Commander J. W. Danenhower, a member of the Nautilus Expedition under Sir Hubert Wilkins. From these one can visualize in part the role of the United States Navy in Arctic Exploration up to World War II.

During the late war, a number of operations, especially in Northern Greenland waters and particularly those conducted by survey ships, must be considered both on the basis of their intrepidity and scientific achievement as creditable missions in Arctic exploration. Among others was the young Naval officer, David C. Nutt, Commander, U.S.N.R., who was with Captain Bob Bartlett during the cruises of the *Effie Morrissey* from 1935 to 1940. On the 1940 cruise he was navigator. During the early years of the war Nutt was engaged in survey work in the Greenland waters.

After the close of World War II, the Navy continued its combined operations and exploration and sent two observers to the Canadian Army winter Arctic exercise, Musk-Ox, 1945-46; Lieutenant Commander R. J. Willingham, U.S.N.R., who was with the base and supply forces and the writer, then Lieutenant Commander U.S.N.R., who traveled with the Moving Force. The Moving Force traveled by motorized tracked vehicles (snowmobiles) from Churchill on the western shores of Hudson Bay 3100 miles across the Canadian Arctic prairies, Queen Maud Gulf, Coronation Gulf and southward from Coppermine to Port Radium across Great Bear Lake down the bush country along the Alaskan-Canadian highway to Edmonton. To compare mechanized traveling techniques with the Eskimo style, the writer and Lieutenant Colonel Frank Forrest, U.S.A., traveled by dog sled from Coppermine, NWT, to Cambridge Bay, Victoria Island, and later rejoined the Moving Forces.

The Naval Operation "Nanook", which had as its primary mission the establishment of Arctic weather stations, must be considered an example of a combined operational and scientific mission which succeeded in both instances.

The experience of the Navy, chiefly of the Bureau of Yards and Docks and the Naval Petroleum Reserve No. 4 in the years

WELLINGTON CHANNEL IN EARLY FALL.

BEACH WEST OF ENTRANCE TO DUNDAS HARBOR.

LOOKING EAST FROM SLOPE OF MOUNTAIN ON SOUTH SIDE OF TOY BAY, BYLOT ISLAND.

LONE STRUCTURE AT WINTER HARBOR, MELVILLE ISLAND, PICTURED IN EARLY FALL.

Along West Shore of Hudson Bay, in February, North of Churchill, Manitoba.

Oil Rig "Go Devil" Mounted on Sled Runners.

MURKY WINTER AFTERNOON, CHURCHILL, MANITOBA.

ALONG CORONATION GULF IN EARLY SPRING SOUTH OF SOME OF THE MANY UNNAMED ISLANDS.

since 1944 while engaged in petroleum exploration in the Point Barrow region, can be considered one of the highest types and the most advanced of scientific exploration of the Arctic. This group has been engaged in actual exploration and has maintained an all year round operating schedule that is not only pioneering, but actually leads the way in establishing a pattern for Arctic colonization. Although many officers of the Navy are interested in this program, the leadership in this work is that of Commodore William G. Greenman.

It is difficult to draw any line of demarcation between the past and the present, for the Navy is now engaged in operations amost routine in nature which, however, must also be considered as exploration.

It may be of value to examine very briefly the historical background of Arctic research in general in order that the picture of the Navy's past and the planned program of the Navy's future in Arctic exploration can be viewed in full prospective.

The discovery of the Arctic, is, as Stefansson has pointed out in his *Great Adventures and Explorations*, that the Arctic was not discovered, but rediscovered.

As early as the end of the eighth century, Iceland was discovered by the Irish and the Irish Monk, Dicuil, described in *DeMensura Orbis Terrae* (820 A. D.) the voyage of some of his countrymen to Iceland. As early as 983 A. D., when Eric the Red was driven out of the country because of a blood feud, he sailed on the famous voyage which resulted in the discovery of Greenland and establishment of a large settlement. During the fourteenth century the Greenland Colony, estimated by Professor Finne Jonsson of the University of Copenhagen, about 10,000 people, was slowly cut off from essential supplies from the mainland and the survivors probably absorbed by the native Greenlanders. The history of the close of the first period of Arctic exploration remains a mystery.

In the fifteenth and sixteenth centuries the development of sea trade to the Far East by the ambitious maritime powers of Europe was the most compelling motive of exploration. This search for new trade routes to the Indies led the English and Dutch to look for a passage across the Arctic seas to the Northern part of Asia and America. In 1845 Sir John Franklin in command of two ships, the *Erebus* and the *Terror*, manned with one hundred and twenty-nine officers and men from England's most

prominent families, sailed in search of the Northwest passage. None returned and in spite of an official search, which lasted for more than eleven years after the expedition was last heard from, no living survivor was found. This tragedy stifled further Arctic exploration for commercial ends.

Toward the turn of the century, exploration of the North was revived. Now, however, the motive was more akin to sport and conquering the polar regions was like climbing mountains. The participants became inclined to stress hardships and dangers and to make spirited dashes to achieve certain geographical places. Scientific observation was, in general, secondary. However, this was not always the case, for late in the 19th century and early in the 20th century the scientific explorations of Nansen, Sverdrup, Nordenskjold, Rasmussen, and Stefansson and certain others were ones of greater scientific accomplishment. Actually these men reoriented the polar exploration into its present pattern of scientific study. Today few men go into the North without some technical interest. This is of significance in the long-time planning and execution of Arctic research and will be discussed later.

The increasing interest in the Arctic has revealed an acute absence of any systematic body of scientific data about the basic geographical and physical conditions. The information which has been gathered in the past century has been chiefly through the toil of individuals or small expeditionary groups whose main mission was generally the attainment of some specific geographic location, and thus the technical information was secured as a secondary objective. While this was not universally true, even on those expeditions with a singular mission the accumulation of scientific data was usually limited both technically and financially, and the parties were usually in the field relatively short periods (expeditions by Stefansson, Nansen, Sverdrup, Koch and Rasmussen were exceptions).

The advances in technology of scientific research are at present of such magnitude that studies must have full financial support in order to exploit the most modern and highly specialized techniques and to procure the greatest benefits from these studies. This demands financial support of a magnitude that government funds are almost mandatory. Moreover, the accumulation of systematically collected data demands long-term studies at fixed

or semi-fixed bases and this again necessitates adequate financial support.

Another important aspect of Arctic research which must be borne in mind in planning is the essential need for the integration of the various disciplines of science. The polar regions are fertile frontiers for investigation especially of the descriptive phases of the natural and biological sciences. But more than that, the polar regions demand the integration of man's capabilities and limitations with factors of engineering design and development, of climate and geography. These polar areas demand a continuing concern by the engineer, the economist, the geographer, and the naval officer of man's reactions to the environment and a continuing regard for the geographic parameters of physical and biological factors by the biologist, the engineer, and the planner, whether they be civilian or military strategists. In an article entitled, *Research in Problems of Human Biology in Polar and Circumpolar Areas*, M. C. Shelesnyak states "it should be noted that problems are included which deal specifically with physical factors of the environment. The study and understanding of these are essential for the investigation of man's reactions to the environment. Their inclusion in this biological program emphasizes the need for close integration of all fields of study when analysis of the polar region is made". Because of this, it is essential to continue collateral and integrated physical and biological studies in Arctic regions. This demands large, highly organized, fully supported, research teams and expeditions.

The Navy's interest in the Arctic has become increasingly important with the greater understanding of the Arctic regions and conditions. In the first instance the fact that the Polar Basin is essentially seas and although a broken layer of ice exists in many areas the waters are deep, the ice moves and there are many leads (open cracks ranging from several feet to miles in width and from yards to miles in length) cannot be neglected. In certain seasons and in certain areas surface navigation is feasible. According to Sub-Inspector Henry A. Larsen, the Captain of the Royal Canadian Mounted Police Schooner *St. Roch* which has made the Northwest passage both from East to West and West to East, the Southern waters of the Canadian Arctic Archipelago along the mainland coast can be navigated almost any season.

Moreover, the utilization of sub-surface vessels under pack ice

should be pursued. This is important particularly in regard to an especially modified submersible for conducting scientific investigations in the Arctic Sea.

A second factor which relates the U. S. Navy to the Arctic regions is the existence of Naval aviation. Long since 1922 when Stefansson in an article in the National Geographic Magazine presented the significance of the great circle courses over the North Pole in the light of aviation future, the Arctic has become important to the flying world.

A third factor which leads the U. S. Navy to interest itself in the Arctic is the existence of ground forces within the Navy organization, as exemplified by maintenance bases and by the United States Marine Corps. With such components a part of the integral structure of the Navy, it becomes vital that the complete understanding of the land and sea of the Arctic and Subarctic be well understood.

The Office of Naval Research was temporarily established as the Office of Research and Inventions in 1945, by order of the Secretary of the Navy, and permanently established the Office of Naval Research by Act of Congress (Public Law 588) approved 1 August 1946. The Office of Naval Research has operated under certain guiding principles to enable it to carry out its chartered responsibilities for the encouragement and sponsorship of basic scientific research to serve the needs of the Navy and the Nation.

A primary responsibility, research, embraces stimulation and coordination of Navy research, and the arrangement of contractual relationships in the field of research with individual civilian agencies and institutions. The Office of Naval Research further functions to stimulate and encourage collaboration with civilian scientists in the sponsorship of fundamental scientific investigations. Research contracts with individuals, private research institutions, and universities for basic research and consulting services are in the nature of partnership agreements and offer the scientist freedom to initiate, explore, test, and publish in carrying out basic research for the Navy under contract.

In accordance with this basic policy a broad program relative to Arctic research has been established and certain phases of this program initiated. In the summer of 1946, the initiation of the Arctic program was accomplished through the Medical Sciences Division by the branch concerned with the environmental factors

as related to man, the Environmental Biology Branch. In order to broaden this activity, an Arctic-Tropic Panel which was advisory in nature and which consisted of interested members from various branches within the Office of Naval Research was established. Plans are under way for the establishment in the near future of a Geographical Exploration Branch within the Naval Sciences Division of the Office of Naval Research which will have as its primary concern the encouragement and sponsorship of geographical sciences.

One of the initial programs to be undertaken in the Arctic program was the establishment of an Arctic Scientific Station at Point Barrow, Alaska. On 13 August the Office of Naval Research (then the Office of Research and Inventions) addressed an inquiry to all Bureaus relative to the establishment of a basic scientific research laboratory in the Arctic, in the vicinity of Point Barrow, Alaska, where civilian scientists under contract to the Navy Department would conduct investigations of physical and biological phenomena related to the environment. In conformance with the general policy of the Office of Naval Research to seek counsel of qualified scientists, well-known Arctic specialists including Sir Hubert Wilkins, Doctor Vilhjalmur Stefansson, Doctor Laurence Gould, President of Carleton College and Geologist to the Byrd Expedition, Doctor A. D. Washburn, Director of the Arctic Institute of North America, Doctor H. U. Sverdrup, Polar Explorer and Head of Scripps Oceanographic Institute, Doctor Paul Siple, War Department, and others, were solicited for their opinions. All expressed a keen interest in the program for the establishment of an Arctic Scientific Station and an enthusiastic appreciation of the Navy's assumption of leadership in this undeveloped field of endeavor.

The nature of the comments of these civilian specialists can be best indicated by excerpts from their replies.

Dr. V. Stefansson replied in part, "The program outlined in your letter of 24 September is so important that I would like to do what you ask in your last paragraph. But the views you asked for would come under most of the heads you mentioned; and since these things have been a life work with me, the comments you invite would fill a book" . . . "Summing up: the program you outline seems to me important, and I would be glad to collaborate both personally and through our library staff."

Dr. A. L. Washburn, Director of the Arctic Institute of America,

wrote: "Your letter of September 24th outlines work of the greatest value to Arctic research and I am pleased indeed to have this opportunity to follow your suggestion and comment on the proposed program. An Arctic field laboratory for the pursuit of basic scientific research is one of the best ways to facilitate the acquirement of new information. It affords a base of operations for scientists, encourages cooperative observations in different fields of science, and is the only means of effecting some types of research requiring frequent observations at fixed points over a period of years. Much of the knowledge that has been gained about the Soviet Arctic was made possible by the establishment of permanently staffed research stations. The idea of such stations is also favored in Canada where the Department of Mines and Resources has already started work on a research station at Baker Lake, 400 miles north of Churchill. The Arctic Institute is keenly interested in all the fields of endeavor outlined by your letter and will be glad to cooperate in every way it can".

Sir Hubert Wilkins, one of the outstanding living explorers, who was a consultant with the War Department wrote as follows: "The information that the Office of Naval Research is effecting research and development in various fields of the basic sciences by contractural agreements with various civilian agencies as well as within Naval laboratories is most encouraging and will certainly lead to results of high value.

"In the formation of programs in specific fields for research which are of urgent importance to the Services, it will no doubt be wise to consider the possibility of grouping teams of workers who can work coordinately toward specific ends and in such manner as to expedite overall accomplishments and avoid duplication of effort.

"In regard to such effort in relation to matter pertaining to the Arctic Regions, one of the primary requirements is the organization of the data already on record and which is presently scattered and probably inconspicuous because of its incidental association with other matters in various publications. In view of this, research and writing of an encyclopedia of the Arctic, a compilation of a bibliographic index of Arctic literature, and cross indexing should be undertaken without delay.

"While the forementioned operations are, in my opinion, of primary importance, the initiation of a service laboratory and field research should not be delayed, since there are specific service

problems in physiology, environmental protection and geophysics which must be solved as early as possible if the services are to be adequately prepared for contingencies evidenced.

"It would not be possible even to outline, in the compass of this letter, the various phases of development necessary, but I might mention as one, the development of suitable seas and land travel in high latitude; some of which might be brought about by the modification of existing types."

It was envisioned that the initiation program of such a laboratory would be focused around specific research projects submitted by leading scientists desiring to do work in the field, rather than around the development of a large physical plant. Through the cooperation of the Bureau of Yards and Docks it was possible to plan for the billeting of scientific groups at the Barrow Station. After the completion of preliminary investigations regarding the type of basic research which would interest the various Bureaus within the Navy, projects which included complete study of climatic conditions, extensive geographical and hydrographical surveys and biological evaluations including medical aspects of the region, an all-Navy conference was held in January, 1947, and final plans made for the launching of the program.

By the summer of 1947 two teams of scientists, one from Swarthmore College and one from Cornell University were under way conducting a year's study on the biological aspects of the Arctic, especially those related to the metabolism of warm blooded animals including man. The establishment of this scientific station by the Navy is the signal achievement and is the first laboratory to be established in the North American Arctic for the single purpose of pursuing basic scientific investigations. Matters related to development and test, although of vital importance are not included in this program.

A second phase of the Arctic program of the Office of Naval Research is related to the existing information, and is in the nature of library research. The traditional writings relative to the Arctic are general in nature, and special items of scientific values are usually included and buried in travelogs, on trip and voyage reports, so that the extraction of specific information demands an advanced student of Arctic literature.

In order that the Nation's scientists and the Navy, in particular, may have ready access to a systematic compilation of all known Arctic information, the Office of Naval Research is spon-

soring the famous student and explorer of the Arctic, Doctor Vilhjalmur Stefansson, in the compilation, editing, and writing of an encyclopedia of the Arctic. The staff now compiling the Encyclopedia Arctica at the Stefansson Library, 67 Morton Street, New York, N. Y., consists of eight permanent members and five free lance assistants in addition to an advisory editor and the Canadian editor, Richard Finnie, of Carp, Ontario. In broad outline the Encyclopedia falls into five main categories: biography, geography, government departments, scientific institutions, and public articles. The biographical material will include exploration histories and sketches of men who by their writings or scientific achievement influenced the exploration and development of the northern regions. Geographical descriptions will deal with the land and sea areas, settlements, rivers, cities, together with resources and industrial development, places described, transportation, and so forth. In the section, "Government Departments", historical sketches will be included of all U. S. Government departments and other countries that deal with the Arctic. Similarly, sketches of scientific institutions will describe development of the Arctic.

The public articles will deal with a wide variety of scientific information: anthropology, botany, zoology, geology, medicine, historical problems, practical methods of living and operating, permafrost, equipment, engineering, and others.

Arrangements for the above work fall under three headings: work done in the library or under the immediate supervision of the editor; the work done by special contributors; and that prepared by Government departments or by scientific institutions.

By June 1947, a few months after the contract was let, almost one hundred completed biographical sketches have been received. Detailed geographical information is being collected and compiled, but the actual writing is being delayed until more material is available. Governments other than that of the United States will cooperate by submitting reports of their activities. The scientific institutions so far have responded with great interest and willingness to furnish articles regarding their roles in furthering exploration in the Arctic. Special articles are being written, and by the end of this year articles on airport construction on frozen ground, technique of cold weather forest camping, sea-ice camping, and a number of other specific subjects should be com-

pleted. The program has been accelerated from month to month since its inception at the beginning of 1947.

Another project in library research is a task which is being supported jointly by the Navy through the Office of Naval Research and the Army with the Arctic Institute of North America for the compilation of a thoroughly cross indexed bibliography of all Arctic literature. This bibliography will be of great value both to the uninitiated and to the specialist in Arctic studies as a source for quick access to the literature. It will be so arranged that fields, specialists and geographical areas can be approached with ease, and the initial steps of further study and investigation become greatly simplified. The use of a bibliography as a research tool is universal. The lack of a complete Arctic bibliography at present is a serious handicap for persons interested in pursuing Arctic studies.

Somewhat similar to the bibliography of the Arctic is the compilation of a roster of Arctic specialists in the nature of a "Who's Who in the Arctic". This, too, is an undertaking by the Arctic Institute of North America under joint sponsorship by the Army and the Navy and under the direction of the Office of Naval Research. Such a roster will include men and women of various specialties, scientists, frontiersmen, explorers, and students of the Arctic. In addition to its intrinsic value as a roster for ready access to the specialists in the field, it will afford an opportunity for students to seek advice and information from various people whose experience and knowledge of particular subjects or regions in the Arctic are unique and often otherwise unobtainable.

In addition to the sponsorship of an Arctic research laboratory and the library type of research noted above, the Office of Naval Research has sponsored, through the Arctic Institute of North America six Arctic field projects, three related to geological work and three to biological investigations during the summer season of 1947. The nature of this work ranged from fundamental studies on genetic developments of certain species of animals which inhabit parts of the world ranging from the Tropics to the Arctic, to topographical studies correlating the surface characteristics with the permanently frozen gound.

Two projects which are in the very earliest of the planning stages warrant discussion. This is true in spite of the fact that certainty regarding the fulfillment or even the possible achieve-

ment of these problems is indefinite. Both programs involve a great deal of preliminary planning and discussion. The first program is in reference to an Arctic floating research station, namely, an Arctic vessel designed and operated for the essential purpose of conducting scientific investigations in the Arctic seas. Such a vessel should be considered a research facility and designed for minimum operational and maintenance forces and with maximum laboratory and research space. It is interesting to note in passing that in spite of the many vessels that have gone into the Arctic Sea for the purpose of studying natural phenomena, no vessel has yet been designed and built specifically for this purpose. The basic planning of the characteristics of such a vessel is being undertaken at present in the Office of Naval Research by Commander David C. Nutt, in the Geographical Exploration Branch of the Naval Sciences Division.

The other program of this nature is the consideration of an especially designed submarine for conducting scientific investigations under the ice of the polar Mediterranean Sea. This is a long-range program, and the primary role of the Office of Naval Research is the establishment of criteria for a research submarine and the thorough examination of the needs and requirements for such a research facility. Only upon examination of the experiences of the past and the requirements for the future, can this submersible Arctic Research Station be considered.

This program which has been defined as current within the Office of Naval Research's planning has been stated in as specific terms as can be characterized by projects either in effect or under consideration. In more general terms the program for polar exploration and research which is of interest to the Navy can be visualized in two significant publications: the first published in 1928 by the American Geographical Society entitled *Problems of Polar Research*, and the second, Bulletin No. 1 of the Arctic Institute of North America published in March 1946, *A Program of Desirable Scientific Investigations in Arctic North America*, which are statements of scientific needs as expressed by students of the Arctic. As great as are the general interests of Arctic students, so great and extensive are the interests of the Office of Naval Research.

In this statement the author has attempted to indicate that, through the sponsorship and support of basic research in the Arctic, and with the freedom to publish scientific findings, the

Navy is in a manner maintaining and perpetuating the heroic traditions of DeHaven, Kane, DeLong, Wilkes, MacMillan, Byrd, and other Naval Officers in Polar Exploration.

THE EXAGGERATED FIRE HAZARD OF MAGNESIUM.

By Captain Carl J. Lamb, U.S.N.R., (Member).

As was true in the case of aluminum, the price of magnesium in the early days of its development was high and its uses were limited. As one of the earliest uses of aluminum was in thermite welding, one of the earliest uses of magnesium was in flash bulbs, and its most widely publicized use, brought forcefully to the attention of the world, was in incendiary bombs during the early part of World War II.

Very early in the history of both metals they graduated from the field of fire into those fields where their properties of light weight combined with high strength made them extremely desirable. Due to the fact that Dr. Willard Dow pioneered and developed magnesium long after aluminum was widely accepted as structural metal, it has not yet found such wide use, although the wide employment of magnesium in commercial and military engines and planes made possible the efficient and long range operation of fighters, bombers and transport planes.

Because magnesium has strength characteristics comparable to aluminum, but weighs only two-thirds as much, its greatest use is still in aviation. Like other metals, its finely divided particles produced by machine operations may become ignited, but since it is considerably easier, quicker and cheaper to machine than any other metal, a piece of comparable size produces more magnesium particles in a given period of time. However, the aircraft and engine manufacturers easily developed the safe and simple procedures necessary to prevent fires when machining magnesium.

As we all know, the finely divided particles of many other elements and materials may also be ignited. Very disastrous explosions and resultant fires have occurred, to mention a few cases, in flour mills, in boiler rooms burning pulverized coal and in wood working plants due to sawdust.

When alternating current electric generators were first developed by Westinghouse, men of those times considered such current very dangerous. As was found to be the case with

electric fires, water should not be used to fight magnesium fires, when they occur.

It is more than likely that a large part of the belief held by quite a few people, to the effect that magnesium burns easily and is therefore dangerous, is due to the use of magnesium in incendiary bombs in the early days of the recent war. The gasoline jelly incendiary bombs were found to be far more efficient, and as a result gasoline jelly replaced magnesium as incendiary material. Industry and the public use gasoline daily, in tremendous quantities, because the oil refiners have developed safe methods of handling it.

So too, the magnesium industry has developed simple techniques to handle magnesium safely, in any of its forms. Actually magnesium is far safer, from the viewpoint of possible fire hazard than is gasoline. Like practically all other solid materials, magnesium and its alloys must be heated to their ignition points before they will ignite.

As noted, magnesium alloys have been widely used in the construction of aircraft engines for many years. About eight years ago, the Sharples Corporation built ultra-centrifuge bowls of forged magnesium alloy, operating at speeds of 80,000 RPM. Today, magnesium alloys are used in waffle irons, ash trays, Diesel and gasoline engines, and in many other applications where they are subject to elevated temperatures.

Due to its light weight, because it resists shock and vibration better than other structural metals, because it is easier to machine than any other known metal or alloy, and because there is an inexhaustible supply available, magnesium is destined to find ever wider use in fields other than aircraft.

On February 18, 19 and 20 at Wright Field, Dayton, Ohio, an unusual exposition took place under the joint sponsorship of the Air Materiel Command, Army Air Forces and the Magnesium Association. In addition to the Heli-Arc Welding demonstration, for the first time in history magnesium was die cast before the public by Hydropress, Inc., of New York, N. Y. The daily operation of the die casting machine, using melted magnesium alloy, before thousands of people, demonstrated that the melting and die casting of magnesium is simple, practical and efficient when performed in accordance with developed practice and when using proper equipment.

As indicated above, the possibility of fire has been grossly exaggerated. We know magnesium and its alloys must be heated to the melting point before they will burn. Finely divided particles or powder will burn quietly if dry or oil soaked. Conventional liquid extinguishers or CO_2 should never be used. In an emergency, water may be used to limit or confine the fire area.

Magnesium fires may occur, primarily because of ignorance or carelessness, as is true in the case of practically all burnable materials. G-1 Pyrene powder quickly extinguishes magnesium fires. Dry sand is effective in the case of small fires, and the U. S. Navy has had considerable success in extinguishing experimental magnesium fires with Freon gas.

Grindings, turnings and borings from machine tools and floors should be removed at regular and short intervals, and stored in dry, covered metal containers. Only oil-type cutting fluids and coolants should be used when machining magnesium and its alloys.

As a matter of fact, magnesium alloys may play a real part in reducing fire hazard in certain industries. Since they are non-sparking, tools made from them will eliminate dangers existing where steel tools are used at present, especially in the explosive, chemical oil refining, and dry-cleaning industries, or wherever volatile liquid or gas is present.

We are becoming aware of present serious shortages of most metals, and of our increasing dependence upon import to obtain many of them. Due to the inexhaustible domestic supply of magnesium, this metal will necessarily replace the scarce ones, in addition to being widely used because of its light weight and unusual ability to withstand shock and vibration. It is believed that the Air Materiel Command, Army Air Forces, and the Magnesium Association have performed a constructive and educational service to both the armed services and industry in sponsoring an exhibition in which the daily and casual die casting of magnesium alloy in the Hydropress machine was witnessed by thousands of people. This demonstration proved effectively that the previously held fallacies regarding the possibilities of magnesium fires are figments of the imagination today.

DISCUSSION OF THE PAPER. "A STUDY OF THE CAUSE OF HARD SLAG ON FIRESIDES OF NAVAL BOILERS".

By Commander L. C. McCloskey, U.S.N.R.

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Commander McCloskey has contributed to the Journal a paper, for which he and the authority who authorized the study involved deserve great credit. An experienced steam and combustion Engineer can only agree with the major contents and conclusions of Commander McCloskey's paper. There are, however, a couple of points which may be amplified constructively. The writer served aboard three Cruisers and one C.V.E., and by paying close attention to critical points was able to prevent slag deposits, scale or soot formation on the fireside tubes of any of these ships. A very important point, and one which deserves great emphasis when burning oil aboard ship, is to make sure that the oil is dry and free of any moisture, particularly since said moisture contained in the oil bunkers aboard ship will almost certainly be sea water which came aboard with the fuel from the tanker, or which remained in the fuel oil tanks as a result of previous ballasting. In order to keep oil dry it was found necessary to strip all fuel oil tanks of water immediately after fueling, to test all tanks for water every 24 hours, stripping same of water when found, and to test and strip the daily fuel oil service tanks each time they were filled. Such procedure may seem irksome but it is very much worth while and pays real dividends—not only in elimination of emergency shut-downs due to losing fire under boilers at sea, but also due to elimination of the sea salts which contaminate fuel oil and which are major contributing factors toward slag formation.

As Commander McCloskey states—fuel oil is a high ignition, high viscosity, tar, held in a solution by a thin petroleum dis-

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tillate. It was originally specified and blended to be atomized at 150 SSU. It is felt that Commander McCloskey was not sufficiently accurate or definite when he stated that viscosity of 150 SSU should result when the oil is heated to within 2 or 3 degrees of 135 degrees F.

The original and pioneering work on proper fuel oil combustion was done by Ralph Brierly at the Boiler and Turbine Laboratory, Philadelphia Navy Ship Yard when it was operated as the Fuel Oil School back in the early 1920's. Mr. Brierly at that time established for the benefit of the Navy, The Shipping Board, the Coast Guard and the oil companies the definite data that the ideal atomization and combustion of fuel oil requires that the oil be heated to that temperature which will result in a viscosity of 150 SSU. Since Navy specification fuel oils are blended of residues of practically all known crude stocks, with distillates obtained from crudes originating in many fields, they are not standard as to the temperature which will give a resultant viscosity of 150 SSU.

With the improvement of boiler practice over the years, and the addition of such heat exchange elements as water walls, superheaters and economizers to Naval boilers, stack temperatures have dropped appreciably, resulting in appreciably reduced gas volumes and velocities. As might be expected, modern boilers have thus become soot and ash collectors. Based upon the above facts, the writer operated Naval boilers so as to have between 10 to 15° excess air. Such operation resulted in keeping all tubes so clean, with the oil temperature correct, that it was only necessary to blow tubes once a week instead of three times a day. As presented before the semi-annual meeting of the A.S.M.E. in June, at Chicago by the A.S.M.E. special research committee on furnace performance, tests on a 475,000 lb. of steam per hr. boiler, 1375 psig, 925° F at the superheater outlet, demonstrated that the use of 15 per cent excess air results in great reduction of ash accumulation (slag), reduced stack temperature and increased absorption rates, due to tubes being clean. From the above it is evident that to operate with a "light brown haze" is not only not good practice: it is definitely bad practice.

In the writer's four years experience at sea in combatant ships, during World War II, fuel oil was only burned at the temperature (as found by taking the actual viscosity of the oil at every fuel-

ing) which would give a viscosity of 150 SSU at the sprayer plate. These temperatures ranged during the four years, from a minimum of 110° F to a maximum of 125° F, but in no case where the oil was obtained from a tanker, a U. S. Port, in an advanced base, or in a rear area base was the oil received aboard which required heating above 125° to achieve the proper viscosity of 150 SSU.

With respect to the pressure range of the so called "semi-wide range" burner it was soon found aboard ship that the lower limit of atomization pressure was too low at 75 psig. At a lower limit of 125 psig and an upper limit of 350 psig on the U.S.S. *Atlanta* excellent results were obtained. Incidentally, when the lower limit is much below 125 psig it requires excessive air to prevent smoking, so that the plant will tell the alert operating Engineer what is wrong. As the author states, the throttling of air registers is bad and defeats the purpose of the design Engineer. It should be self evident that in every case best results will be obtained when operating with a maximum number of small size burners at high pressure rather than only a few large size burners at a reduced pressure.

The writer can not agree entirely with Commander McCloskey's "Means for Reducing the Incidence of Slag Deposits" on pages 161 and 162 and offers comments as follows:

A. *Modification of oil design.* In view of the present World shortage of mineral oil deposits and the fact that the World demand exceeds the supply, it will be impossible in the case of future wars to obtain "ideal" or improved fuel oils. As a matter of fact when the Navy, Army, Air Forces and industrial demands for lubricants and gasoline are considered it will be doubtful if the Navy will be able to obtain oil anywhere near the high quality of that burned in World War II.

B. *Modification of Boiler Design.* The use of wider tube pitch will mean larger, heavier boilers and the use of lower superheat temperatures will again mean larger, heavier boilers, turbines and steam piping, in addition to increased fuel consumption.

C. *Modification of Furnace Design.* It is believed that there is room for improvement of furnace shape, not only to give larger flame travel but to eliminate "cold corners". The writer will be very much surprised if work along this line

is not being considered as part of the research program at the present time.

D. *Modification of Burner Design.* It is believed that the Navy, Merchant Service and industrial plants may profit as a result of some of the recent outstanding burner developments, accomplished by such organizations as Allis-Chalmers, Westinghouse, General Electric Company and Bituminous Coal Research, Inc. in the design and construction of atomizer burners for jet engines and gas turbines. It is felt that the use of a large number of small burners will have certain draw backs both from the weight viewpoint and the viewpoint of maneuverability. The "semi-wide range" burner does have some very great military advantages during war time, particularly in large formations which are maneuvering at sea on various courses and at various speeds. With respect to separate double fronts to avoid throttling of air registers—this again would add weight and space and is of course not necessary if the register is operated properly. Ultimate best results would seem to be dependent, as is true with respect to stripping water from oil, and ascertaining the right temperature at which the oil should be burned, upon the knowledge, experience and sense of responsibility of the Engineer Officer.

E. *Operational, Oil.* Sufficient comment has been given above on the importance of stripping water from fuel tanks. At all times the Commanding Officer and the Engineer Officer should insist upon ballasting of all empty fuel oil tanks at sea. When the fuel has been consumed from such tanks the danger involved due to poor ship stability when any Naval vessel is operated at sea with an appreciable number of empty fuel (or fresh water) tanks it too great to ever encourage an operating engineer to reduce the water ballasting of tanks. The destroyers which were lost in a typhoon during World War II did not have their fuel tanks ballasted while sister destroyers, only a few hundred yards away, weathered the same typhoon because their fuel oil tanks were ballasted. The Navy experimented with centrifugal removal of water from bunker oil many years ago. Beyond any question of doubt centrifuges will remove the water from oil—however, the weight and space required are prohibitive for any practical consideration of its use.

G.-H. *Operational, Furnace—Operational, Burner.* The writer agrees with Commander McCloskey with respect to all items under these two headings except 7 and 8 under H. Since the use of a "semi-wide range" burner between 125 psig and 350 psig offers definite and military advantages, and since as is pointed out above, a predetermined fixed oil temperature for obtaining the proper viscosity of 150 SSU can not and should not be given as an arbritary figure, the only proper method of obtaining the desired viscosity of 150 SSU is to find the temperature for each batch of oil received which will give the ideal viscosity by means of the viscosimeter. This is easy since the oil laboratories of Naval vessels all contain viscosimeters as standard equipment.

All Naval Engineers worthy of the name should realize the importance of eliminating all possible water from fuel oil before it is burned. The old Fuel Oil School and the later Boiler Laboratory have taught thousands of officers, water tenders and firemen how to burn fuel oil and for over 20 years have been stressing the importance of ascertaining the right temperature to which the oil should be heated to obtain the ideal viscosity. At the same time the same Navy personnel not only learned through such schools but through the agency of the Manual of Engineering Instructions, the best methods for the proper operation and cleaning of burners, sprayer plates and registers. In the final analysis it is believed that the major part of the slag formation in Naval boilers during wartime was caused mainly by a large percent of inexperienced personnel, who did not know or realize the importance of operating with dry oil, with proper temperature to obtain 150 SSU oil at the sprayer plate, and with a clear furnace at all times. It is the writer's firm opinion that if the Engineer Officer will hold his commissioned and non-commissioned subordinates to a high standard of performance in the operation of modern high pressure, high temperature oil burning boilers the incidence of hard slag deposits on the firesides of Naval boilers will be practically eliminated.

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OPERATION PLUTO.

This article by A. C. Hartley, formerly Technical Director, Petroleum Warefare Department, is reprinted (in abbreviated form) from the Journal and Proceedings of the Institution of Mechanical Engineers for March, 1947. It outlines the difficulties encountered and the various designs tried in developing the cross-channel oil pipe line which supplied gasoline to invasion forces on the continent of Europe.

The code name "Pluto", coined from the initial letters of "Pipe Line Under The Ocean", was chosen for the operation of laying submarine pipe lines across the English Channel and supplying petrol to the Allied Armies of Liberation for their advance across France and Belgium into Germany.

In this lecture I will describe how the urgent operational need for pipe lines came to be clearly stated and how an oil company, a submarine cable company, and a steel company developed, for the Petroleum Warfare Department, two novel methods which, as a result of collaboration between many rival commercial firms, Government Departments, and the Services, were used to deliver more than 1,000,000 gallons of petrol a day across the Channel.

Experiments were made at Westward Ho! where sea conditions were similar to those on the French Coast and, by the beginning of April 1942, Lord Louis Mountbatten became convinced that there would be grave risk of supplies being interrupted by the hazards of weather when tankers were discharging on a lee shore, and that they would be sitting targets for enemy sea and air attack. He therefore decided that a petrol pipe line across the Channel was necessary if supplies were to be assured at all times, and he asked Mr. Geoffrey Lloyd, the Secretary for Petroleum, who was responsible for the English pipe line system, if he could carry that system across to feed the land lines which I have said would be put down by the Royal Engineers as the armies advanced.

I was told of the requirement when I visited the Petroleum Department on 15th April 1942, and at first agreed with the views of the experts, but when the urgent need for a solution of the problem was repeatedly emphasized I suggested the only chance of success would be to make a pipe in one complete length, so as to lay it across without stopping and at sufficient speed to get across the strong tidal currents. This would mean using a small-diameter pipe owing to the bulk and weight, but I then thought of the way in which a difficult pumping problem through hilly country in Iran had been solved. The unusually small internal diameter of 3 inches had been chosen; and by working at the very high pressure of 1500 lb. per sq. in., more than 100,000 gallons, or the equivalent of 25,000 Jerricans a day were being delivered a distance of 40 miles from pumping station to pumping station. I realized how valuable such a line would be and that, if one, many could be laid.

I was immediately encouraged to develop my idea, and I thought of submarine telephone and power cable practice.

Sir William Fraser, C.B.E., Chairman of the Anglo-Iranian Oil Company, Ltd., and Honorary Petroleum Adviser to the War Office, that night offered the fullest support, and I saw Dr. H. R. Wright, Managing Director of Messrs. Siemens Brothers and Company, Ltd., the next morning. He received my idea with enthusiasm and at once agreed to make a trial length of cable armored with steel tape and wire, but without the communicating wires, and conducting insulation, and it would start with the usual lead covering, as an inner lead pipe impervious to petrol. It would be prevented from bursting by the steel tapes and would be held together longitudinally by the steel wires.

Even at this early stage it was realized that secrecy would be essential and it was decided to use the word cable rather than pipe, and the code name Hais was chosen.

The first 200 yards were ready for test in a week and withstood test pressures of up to 700 lb. per sq. in. before failing at the plumbed ends. It was only 2 inches in diameter and designed for a working pressure of 500 lb. per sq. in. It would only deliver about 30,000 gallons a day across the 20 nautical miles then visualized, but when it had been handled in and out of a Post Office Cable Ship and again been tested satisfactorily, samples were shown by

Mr. Lloyd to the Service Chiefs and to Mr. Churchill, then Prime Minister. Instructions were received to proceed with its development with all speed.

The Post Office, the Admiralty, Combined Operations, the War Office, and the Anglo-Iranian Oil Company were called together at the Petroleum Division to arrange the manufacture of further lengths and prepare a complete test program. The Anglo-Iranian then undertook, as agents of the Petroleum Division, to develop, order, progress, and supervise the whole of the pipe, pipe joints, pumping installations, etc., required. Messrs. Siemens Brothers, without waiting for official orders or priorities, quickly produced more cable. A length was laid by the Post Office Telegraph Ship *Alert* as a loop in the Medway at Chatham, on 10th May 1942, and a pumping test was started with pumps borrowed from the Manchester Ship Canal Company, which they were keeping for the emergency operation of lock gates in case of "blitz" damage.

Failures occurred on this length after two days' pumping, and the faulty portions were recovered and examined by the Post Office, by Siemens, and by Henleys, who had, at Messrs. Siemens' suggestion, been brought in to provide more manufacturing capacity. It was soon evident that the chief reason for the failure was the extrusion of the lead through gaps which had been left in the spiral steel tape armoring due to the upper layers of tape having come back directly over the lower instead of breaking joints in certain places along the cable.

A striking example of the way in which the various interests combined all their resources even at this stage to get over difficulties was then provided by Siemens and Henleys combining their Research and Design facilities and together with the Post Office and the National Physical Laboratory, who had also been brought in, helping us to prepare a new specification within two days of the failure. Further lengths of 2-inch (internal) diameter cable were ordered from both cable makers with four layers of steel tape instead of two, to reduce risk of lead extrusion and to enable a higher working pressure to be used.

This was calculated to be 750 lb. per sq. in., on the basis that, allowing for the gaps in the spirals, only three out of the four tapes would be effective and that each would be stressed to half the yield point of the steel.

The lays of the tapes and of the wires were designed for right- and left-hand lay respectively to balance each other so that the cable would not twist under the influence of internal pressure.

Differences in the method of manufacture of lead sheath normally used by the various cable makers had meanwhile been under discussion, and Messrs. Siemens Brothers took the view that their technique, using a vertical press and involving a longitudinal seam, while entirely satisfactory for extruding lead sheath on ordinary cable, might need some development to make it equally satisfactory for making pipe. Rather than run the slightest risk or dealy, they agreed to use lead made in presses which avoid a longitudinal seam. Messrs. Pirelli's lead, made in their continuous extrusion machine, was tested, but before it could be a lopted, their works were damaged by enemy action. Messrs. Henleys' lead. made in their "Judge" straight-through presses, had already been proved suitable, and this type of press provided all the lead pipe used until the capacity of further cable companies had to be brought in to produce the large quantities of cable eventually required. Lead pipe made both by Messrs. Pirelli's continuous presses and by vertical presses both here and in the U. S. A. was later used with complete success.

Test lengths of both firms' manufacture were laid by the Post Office Telegraph Ship *Iris* in deep water off the Clyde. Messrs. Siemens's length was laid first under the severest possible conditions, being laid over the bow and filled with air under only atmospheric pressure. On test, after recovery from the depth of about 200 feet where the external pressure on the cable was 90 lb. per sq. in., it appeared at first to be leaking because the applied internal test pressure would not remain steady after the cable had been filled with water, and because water appeared in places through the outer jute serving. These places were stripped, and the lead pipe was found to have been pressed in on itself into a kidney shape (Fig. 2) by the external water pressure, and some water was found to have been trapped in the space thus formed between

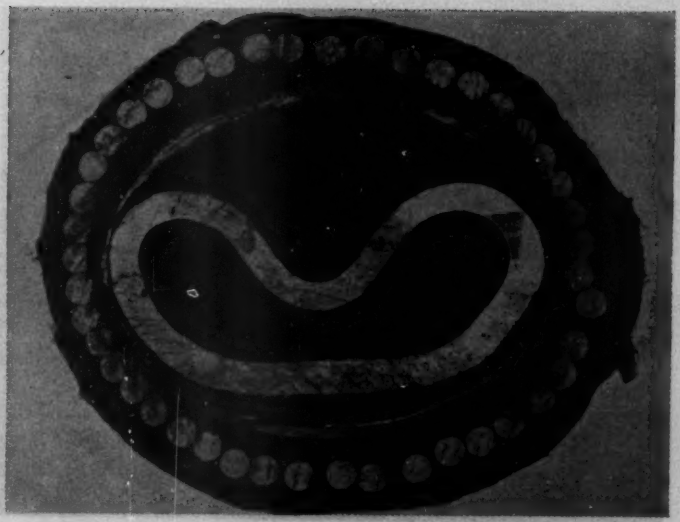


FIG. 2.—KINK PRODUCED IN LEAD PIPE BY EXTERNAL WATER PRESSURE.

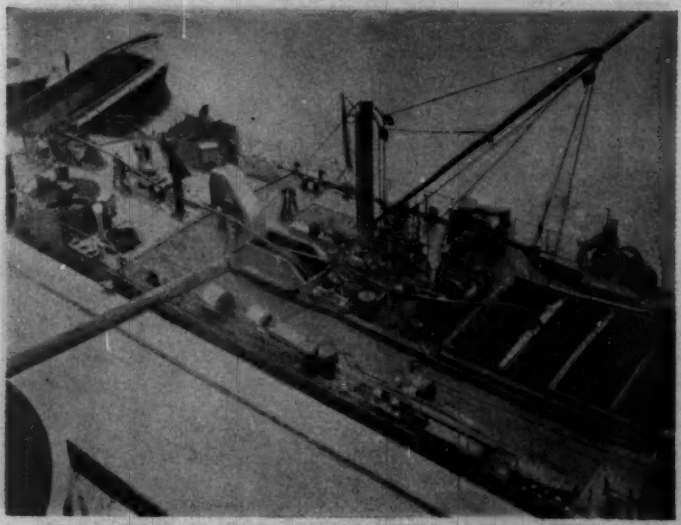


FIG. 3.—LOADING CABLE INTO TANK, H.M.S. "HOLDFAST".

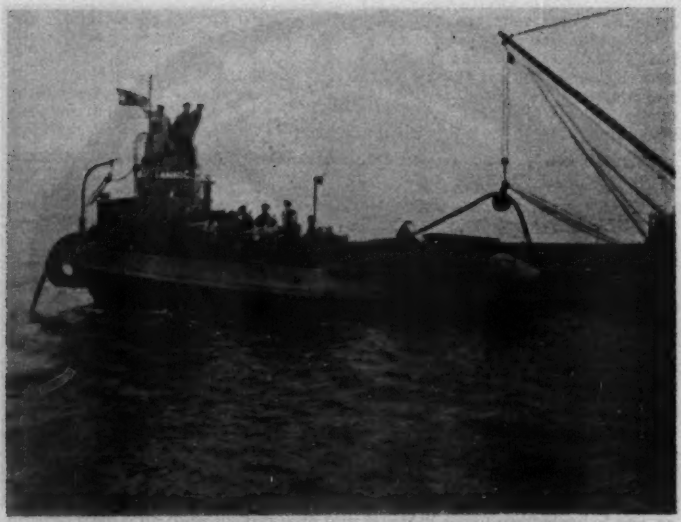


FIG. 5.—MAIN LENGTH BEING CONNECTED TO SHORE END.

FIG. 6.—35-NAUTICAL-MILE LENGTH OF CABLE BEING LOADED FROM FACTORY COILING SITE.

FIG. 7.—STORAGE RACKS FOR HAMEL PIPES.

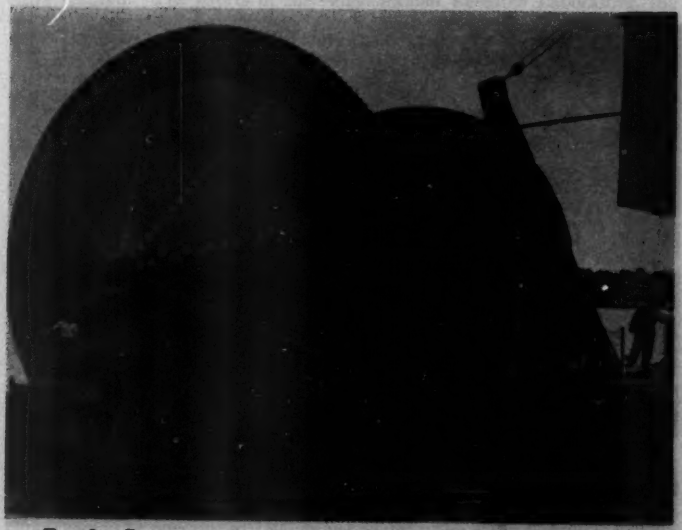


FIG. 8.—CONUN AT JETTY, SHOWING HINGED ARMS FOR HOLDING IT IN POSITION BY ITS TRUNNIONS.

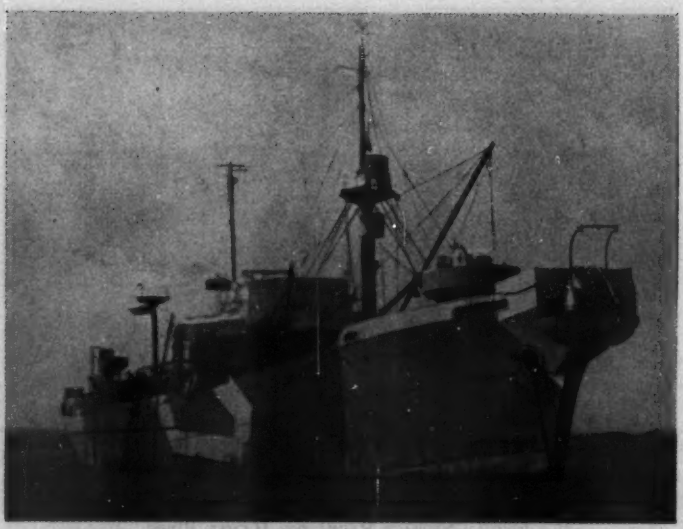


FIG. 9.—H.M.S. "LATIMER", WITH CABLE GEAR.

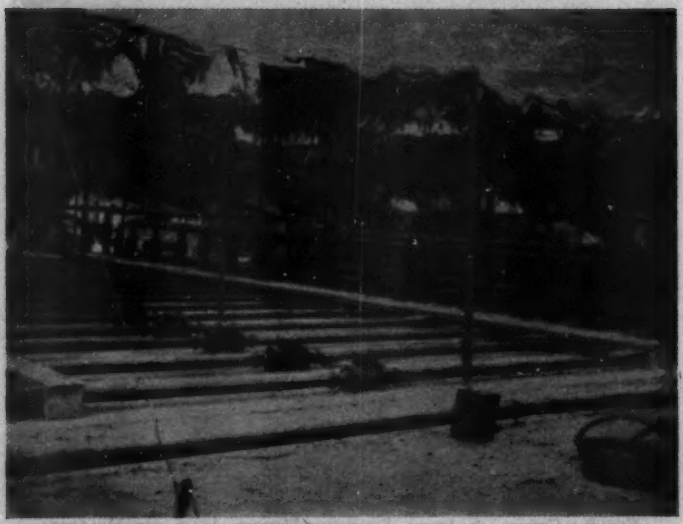


FIG. 10.—VALVE MANIFOLD SYSTEM ON THE BEACH AT BOULOGNE.

the lead pipe and its armoring. Under application of the test pressure, the lead pipe began to resume its circular shape and push the trapped water through the outer armoring, giving the impression of leaks.

Before this explanation had been found, however, the decision had been taken to lay Henleys cable under the easiest possible conditions owing to the increasing urgency for the cross-Channel lines. It was laid over the bow of H. M. T. S. *Iris*, but with the ship going astern to simulate the easier over-the-stern laying condition; and it was laid full of water under 100 lb. per sq. in. inernal pressure, to balance the external sea pressure.

The complete success of this lay and the knowledge that the Siemens cable, far from failing under the severe conditions of its lay, had, in fact, been proved capable of withstanding much rough handling, resulted in the decision to start manufacture of six operational lengths, 2 inches in internal diameter, and of 30 statute miles each, and to use one length for a full-scale trial in the Bristol Channel where conditions of tide and depth of water were more severe than those in the English Channel.

Consideration had meanwhile been given to the methods to be adopted for laying the cable in actual operation, and it was early realized that special cable ships would have to be equipped to carry a sufficient length of this unusually heavy cable, and there still remained the problem of dealing with the shore ends in the shallow water, into which the cable ship could not approach.

The Admiralty and the Ministry of War Transport made available the S. S. *London*, a coaster of 1500 tons, and she was fitted out and renamed H. M. S. *Holdfast* under the direction of the Director of Naval Construction with Johnson and Phillips' cable gear lent by the Post Office.

In the earliest tests on the cable, failures had occurred at or near the fittings provided at the ends for holding the pressure, and the great importance of designing suitable fittings for joining the shore ends to the main lengths was early realized. The efforts of the Anglo-Iranian Research Laboratory, the Admiralty Research Laboratory, and of many commercial firms were later combined and concentrated at Siemens Brothers, and resulted in the preparation of an improved joint and its manufacture by them. Bursting disks of thin copper sheet were incorporated in the joint to hold the water pressure, up to 200 lb. per sq. in., which had been found desirable during handling, storing, and laying, but were so designed as to burst under the operating pressures. Although about three or four hours were required for fitting the joint to the end of the cable, two lengths already fitted could be coupled together at sea in about 20 minutes by using the specially designed split muff coupling without the need for using any bolts. The diameter of the coupling was sufficiently reduced and faired into the cable to allow it to be handled by the cable ship's machinery.

The final design allowed the steel wires and tapes to be gripped securely and for the joints to be made petrol-tight with petrol-resisting rubber-moulded rings against the lead pipe while at the same time preventing the lead from being distorted or extruded. The steel rings locking the two half-muffs together could not be put into position unless the ends of the joints were correctly in line and the muff was fully home, thus making inaccurate assembly impossible.

Static tests on lengths of cable were continued at the makers' works, and pressures in excess of 3000 lb. per sq. in. were maintained for many months. In addition, routine tests, in which a sample of cable was bent without pressure six times round a 6-foot diameter drum, each time in a reversed direction, gave in all cases an average bursting pressure of between 3500 and 4000 lb. per sq. in. It was thus obvious that the fourth tape and also the steel wires, which had been neglected in calculating the working pressure of 750 lb. per sq. in., were in fact contributing largely to the strength; and it was decided that the cables should be operated at 1500 lb. per sq. in. The static tests showed that the lays of the steel tapes and wires were correctly balanced, as there was no tendency for the cable to rotate under the influence of internal pressure until the yield point of the tapes was reached at about 2500 lb. per sq. in.

Throughout the autumn of 1942, Combined Operations conducted handling tests with cable on drums at the experimental establishment at Westward Ho! in an endeavor to find ways of handling the shore ends with craft which would automatically be avialable on the beaches on D-Day. The most promising method devised was to mount two-cable drums with about 1000 yards of cable on horizontal axles in "L. C. T.'s" (landing craft-tanks) with a view to paying the cable out over the bow ramp which was lowered with the craft going astern.

This method was used when the full-scale trial was made on 29th December 1942, starting from Swansea. Although the main length was successfully laid at about 5 knots by H. M. S. *Holdfast* under the command of Commander Treby-Heale, O.B.E., R.N.R., who had been in command of Siemens famous cable-layer *Faraday*, great difficulty was experienced in laying the shore ends owing to the lack of maneuverability of the L. C. T.'s when going astern with heavy cable over the bow.

As a result of a conference at Combined Operations Headquarters in January 1943, it was agreed it would be necessary to adopt the method used commercially of coiling sufficient cable horizontally in the hold of a self-propelling barge specially fitted for paying it out over the stern through hand-controlled compressor gear. Although this involved allotting very precious Thames barges and their crews for this sole purpose, they were made available and the shore ends were completed in a few days' work at the end of March 1943.

The National Oil Refineries at Swansea, with the Royal Engineers, and the Royal Army Service Corps' specially trained Bulk Petroleum Companies, had meanwhile erected a pumping station on the sea wall at Queen's Dock and connected it to their petrol tanks; and the R. E., working with Combined Operations and the R. A. S. C., had, with the help of the Petroleum Board, erected a receiving terminal with tanks, pumps, and loading racks in Watermouth Bay, near Ilfracombe.

After satisfactorily testing with water, the first petrol ever to be pumped through such a long sea line reached Watermouth on 4th April 1943. Mr. Lloyd saw the petrol arriving a few days later, and took a sample to Mr. Churchill.

It had been intended that the vulnerability of the cable to bombing or depth charges and the possibilities of repairs, should it be dragged by a ship's anchor, should be investigated. A German raid on Swansea proved that the cable was not damaged by a bomb within 100 feet, and, during a gale, a ship at the Mumbles anchorage dragged the cable with her anchor. H. M. S. *Holdfast* had no difficulty in locating the cable, cutting out the damaged portion, and running in a new length. We were thus saved, by the enemy and the weather, from the need for making our own experiments!

In order to prove the reliability of the cable and pumps, and to train the R. E. and R. A. S. C. personnel who would be responsible for the actual operation, pumping continued night and day, first at the original design pressure of 750 lb. per sq. in. and later at 1500 lb. per sq. in., when 56,000 gallons were pumped to Watermouth each day and delivered by the Petroleum Board in Devon and Cornwall.

After many months at this rate, the Petroleum Board preferred to receive only about 25,000 gallons a day. It was arranged that the line should continue to be operated at full pressure but only for the time necessary to deliver the required quantity. This meant that during the shut-down periods the line would be under external pressure due to the difference in specific gravity of sea water and petrol, and would therefore be subjected to a daily reversal of pressure. It withstood this severe test with complete success.

The Quartermaster-General to the Forces, Sir Thomas Riddell-Webster, visited Watermouth during Easter 1943, and the cable makers' Works the following week, but before describing the action he took to press the work forward, I will recount the development and manufacture of 3-inch Hais cable and also of the Hamel pipe.

While the manufacture and testing of 2-inch internal diameter cable had been proceeding, experiments were made with larger diameters, and it was proved that one cable 3 inches in internal diameter and 4¼ inches in external

diameter could be made with existing facilities and could be handled by the ship's machinery with only minor modifications. A 3-inch cable would deliver about 2¼ times more petrol than a 2-inch cable, and it was therefore decided to stop the manufacture of the 2-inch and concentrate on the 3-inch. I will now give you the specification and describe the method of manufacture.

Of the 570 nautical miles of lead sheath used in the United Kingdom 463 were made on the Henley straight-through presses by Messrs. Henley and the Telegraph Construction and Maintenance Company, and 87 nautical miles on the Pirelli continuous extrusion machine by Messrs. Pirelli, Johnson and Phillips, the Standard Telephone Company, and the Edison Swan Electric Company, the balance being supplied by Messrs. Glover's on their "Farmer" press. The lead sheath has an internal bore of 3.05 inches with a minimum lead thickness of 0.175 inch, and was manufactured in 700-yard drum lengths. The drums of lead sheath were placed behind the armoring machines and each individual length was joined by lead burning to the continuous length being armored. The cable was armored, and a pressure of 25 lb. per sq. in. was maintained to prevent collapse.

The lead sheath was first coated with a layer of petroleum residue compound, followed by two layers of compounded paper tape with a width of 2½ inches and a thickness of 0.010 inch, applied left-hand lay with a gap of approximately ⅛ inch. This wrapping is then followed by one layer of compounded cotton tape about 3 inches wide, having an overlap of ⅛ inch, these materials being applied to give a bedding for the steel tape and also to increase flexibility. Then follow four layers of steel tape 2 inches wide, with a thickness of 0.022 inch, made from steel with a minimum tensile strength of 25 tons per sq. in., a minimum yield of 20 tons per sq. in., and an elongation of 10 per cent in a length of 10 inches. These steel tapes are applied right-hand lay with a gap of 0.15 inch, breaking joint about "60/40". The individual tapes are joined together by cutting at 30 deg. to the length of the tape and lap spot welding. The diameter of the cable over these tapes is about 3.72 inches.

There then follows a petroleum residue compounded coating with a first serving of single-ply jute bedding, right-hand lay, and a further coating of compound applied at the armoring die, to provide a flexible bedding for the armoring wires. There are 57 galvanized steel wires each 0.92 inch in diameter, having a tensile strength of 25-30 tons per sq. in. with elongation of 12 per cent minimum in a length of 10 inches, and they are applied left-hand at about 30 inches lay, with an approximate diameter over the wires of 4.19 inches. The wires are applied from a rotating carriage having floating bobbins to eliminate twist of the wire, and each individual length is joined by butt welding. A second serving of single-ply jute is put on right-hand lay with a petroleum residue compound over the jute. A third serving of three-ply jute is now added and is applied left-hand lay, after which there is a layer of compound followed by a final coat of whitewash solution.

The finished cable, full of water, weighs 63 tons per nautical mile, and was made in continuous lengths of 35 nautical miles, coiled direct from the armoring machine into special storage sites to await loading on to the ship. Fig. 6, shows a 35-nautical mile length being loaded from the factory coiling site; the diameter of the coil is approximately 60 feet.

At the commencement of the manufacture of 3-inch Hais cable it became necessary to obtain much greater manufacturing capacity; and therefore additional machines were brought into operation at Messrs. Callender's, Glover's, and Pirelli's. In the case of Messrs. Callender's, four machines were finally brought into production, and an overhead gantry 45 feet high by 1600 feet long with supporting towers every 70 feet, was constructed to carry cable from their amoring shop to an adjacent deep-water jetty. Eight 60-foot diameter coiling sites were situated between the supporting towers to facilitate continuous manufacture and loading; and these sites were covered by a continuous light-framed steel building which had to be erected due to blackout regulations. A similar arrangement was made for two amoring machines at Messrs. Glover's and for storage and loading at the Manchester Ship Canal.

In February 1944, I visited the United States to arrange the manufacture of cable at four works there, to meet the increased requirements.

Messrs. Phelps-Dodge erected a plant alongside deep water at Yonkers, New Jersey, in which they installed two armoring machines delivering cables to a specially designed coiling site which had additional facilities for receiving 3½-mile lengths of cable, delivered coiled down on groups of four freight cars from Messrs. General Electric and Okonite-Callenders. These lengths were spliced into continuous lengths at the site. Messrs. General Cables Corporation manufactured a length of cable at their Perth Amboy works which are adjacent to deep water. The cable was delivered to the United Kingdom in merchant ships specially fitted with cable tanks.

A total of 710 nautical miles of Hais cable was produced for Operation Pluto, of which 140 miles came from the United States.

The second novel proposal came at the end of April 1942 from Mr. B. J. Ellis, O.B.E., chief oilfields engineer of the Burmah Oil Company, and Mr. H. A. Hammick, O.B.E., chief engineer of the Iraq Petroleum Company, both of whom were at that time seconded by their Companies to the Petroleum Department. They were dealing with the Hais cable, and when they saw how flexible it was in a long length, although extremely stiff in a short length, they suggested that steel pipe, which they had also seen to be very flexible, when handled in long lengths in the oilfields, might also be used for making long lengths of line required in one piece.

With the assistance of Messrs. Stewarts and Lloyds, Messrs. J. and E. Hall, of Dartford, and Messrs. A. I. Welding, they quickly proved that 3-inch steel pipe could be bent round a wheel 30 feet in diameter, could be pulled off again relatively straight without kinking, and could be flash-welded to provide any required length. They realized that it could not be handled like cable in horizontal stationary coils in a cable ship because this involves a complete twist in each turn while coiling down, and its removal while uncoiling for laying. Mr. Ellis therefore suggested, first, the use of a large wheel mounted on trunnions on the deck of a hopper barge with its lower portion protruding into the sea through the hopper doors, and later a floating drum like a gigantic cotton bobbin capable of carrying any quantity of pipe likely to be required.

Model tests of the floating drum—H. M. S. *Conundrum* or "Conun", as it came to be called—were made at the National Physical Laboratory in their Froude tank and confirmed that such a vessel could be towed at sufficient speed without yawing.

Preliminary work had thus proved that the pipe could be bent and pulled off straight, that it could be welded with absolute reliability, and that it could be carried and laid by either the wheel and barge or the Conuns, but there was no previous experience as to how bare steel pipe would lie and behave on the bottom of the sea. It was felt, however, that it would have at least a six-weeks' life, and on this basis of life, and it was by no means certain there would be sufficient supplies of lead available to produce all the Hais cable required, it was decided to proceed with all speed with a factory at Tilbury to weld, store, and wind Hamel pipe and with the conversion of a hopper barge—later called H. M. S. *Persephone*—and with the construction of six Conuns.

Messrs. Stewarts and Lloyds undertook, in addition to making the steel pipe, to act as agents of the Petroleum Division for the design, construction, and subsequently the operation, of the Tilbury factory; and the Director of Naval Construction fitted out *Persephone*, designed the Conun, and supervised their construction by Messrs. Orthostyle.

Each Conun was 90 feet long, with a diameter of 40 feet and a length of 60 feet between the flanges. Each flange was 52 feet in diameter, having teeth on the periphery for rotating the Conun in the winding operations by a chain drive. It could carry up to 70 miles of 3-inch steel pipe at a time, and weighed with this length of pipe 1600 tons. The hollow steel trunnions, 10¼ inches in diameter, had Timken roller bearings in cylindrical cast steel housings having pairs of vertical trunnion pins 4 inches in diameter, to which two towing lines would be attached. The housings were closed at each end by cast steel covers, each of which carried a special arrangement of grease seals. Each pair of

bearings was designed to withstand a breaking load of 130 tons on the tow rope, as well as a 40-ton thrust load due to the angular pull of the tow ropes.

Two factories with storage sites were erected at Tilbury, each having seven lines of machines consisting of a flash-welding machine, a Taylors pipe-cutter, and a traversing machine. The standard 40-foot lengths of pipe were welded one by one with "A. I" flash-welding machines to form lengths of 4000 feet.

Low-carbon mild steel was used to facilitate the welding, and the pipe was made 3½ inches in external diameter, with a wall thickness of 0.212 inch and a weight of 20.21 tons a nautical mile.

Beads standing about ¼ inch proud of the pipe were produced by the flash-welding process, and they were machined down to about ⅛ inch internally by a rotating cutter and air-blast brushes on long spears, and externally by a Taylor cutter.

The traversing machines had top and bottom tracks which gripped the pipe with spring-loaded pads and pushed it along the 4000-foot long conveyor channels as each new length of pipe was welded on. Storage space was provided below the conveyors, and when each length was completed a few yards were lifted off at one end, and the whole length then threw itself off, due to its weight and elasticity, at about 120 Mph. Fig. 7, shows the storage racks with the winding site in the background.

A welding machine was provided there for joinging the 4000-foot lengths into the continuous length required; and a specially designed hydraulically loaded machine, similar to the snubbing gear used on oil wells, was provided for maintaining the steady back-pull on the pipe of about 2 tons, which was necessary for bending it round the Conun during winding.

Hinged arms were provided at the end of the jetty to hold the Conun in position by its trunnions, and means were provided to ballast it with water when light to minimize wind effect. The pipe was guided on to the Conun through a traversing cab to ensure neat winding; and spring spacers were clipped on to the pipe as it was wound, to maintain passages between adjacent pipes through which water, trapped as the Conun rotated during the lay, could escape. It was thought that without this means of escape, sufficient water might be picked up to cause excessive braking effect and increase the pull necessary during the lay.

Pending completion of the Tilbury factory, some miles of 3-inch steel pipe were hand-welded at Portsmouth Dockyard and wound on *Persephone's* wheel for preliminary trials which were, to the admitted astonishment of most of the spectators, entirely successful. This was early in April 1943, so both the Hais cable and Hamel pipe had been brought successfully through their full-scale trials, and production on a considerable scale had been organized by the Petroleum Division and Chief of Combined Operations before they handed on responsibility for the Operational Stage respectively to the Petroleum Warfare Department, under its Director-General, Major-General Sir Donald Banks, K.C.B., D.S.O., M.C., T.D., and to Force Pluto specially organized by the Admiralty under the command of Captain J. F. Hutchings, C.B.E., D.S.O., R.N.

Thus, after visiting Watermouth on 24th April 1943, and seeing the Hais cable in actual operation, the Quartermaster-General was able, on the 29th April, to visit the Hamel factory at Tilbury, to see Hais cable in production at both Henleys at Gravesend and at Siemens's at Woolwich, where he also saw H. M. S. *Holdfast* loading cable. He decided that further lengths should be ordered at once.

During June and July 1943, recommendations were made by the Quartermaster-General's Petroleum Committee, and confirmed by the Chiefs of Staff Committee, who awarded a high priority, that the English pipe-line system should be extended to Dungeness and to the Isle of Wight and that pumping stations of 3500 and 3000 tons a day capacity, respectively, should be erected at these places.

The Isle of Wight to Cherbourg crossing, then considered for the first time, involved a sea crossing of about 70 nautical miles instead of the 20 or so orginally visualized, and made necessary the provision of larger cable ships

and the use of the Conun loaded down till the axles were awash. Following a successful lay with 3-inch Hais cable, S. N. O. Pluto obtained three more ships to be converted and fitted with cable gear by the Director of Naval Construction, one, H. M. S. *Algerian* to carry 30 miles of 3-inch cable, and the other two, H. M. S. *Latimer* and H. M. S. *Sancroft*, to carry 100 miles of 3-inch cable weighing about 6400 tons. Six Thames barges were equipped to handle the shore ends and a large number of auxiliary vessels were added to the Force, whose personnel numbered 100 officers and 1000 men.

Further tests with a model Conun at the National Physical Laboratory showed it could be handled when loaded with 70 miles of pipe, provided two of the largest class of ocean rescue tugs were used ahead, with a smaller tug astern for steering.

The Isle of Wight pumping installation at Sandown consisted of sixteen reciprocating and two centrifugal pumps, and at Shanklin, eight reciprocating and two centrifugal pumps, these stations being cross-connected by two Hamel line loops laid out to sea and in again. Thus if either of the installations at Sandown or Shanklin had been "blitzed" the other could have taken over.

At Dungeness, thirty reciprocating and four centrifugal pumps, designed for 1500 lb. per sq. in. pressure, were installed at three well-dispersed sites along the coast and were fed from the land-line system.

The reciprocating pumps were manufactured by Messrs. Frank Pearn, Ltd., and delivered approximately 40 Imperial gallons of gasoline per minute, when running at a speed of 45 Rpm. They were driven by V-ropes by 60 Hp. "Caterpillar" engines. The centrifugal pumps were manufactured by Messrs. Mather and Platt, Ltd., and delivered 214 Imperial gallons of gasoline a minute, and were driven by 500 Hp. motors.

The Anglo-Iranian undertook the supervision of the erection of the pumping terminals and tankage by civilian contractors and R. E., R. A. S. C., and Pioneers.

The Petroleum Board constructed the land lines and S. N. O. Pluto laid a large number of Hais and Hamel lines across the Solent, both to provide the link across to the Isle of Wight, and also, at the same time, to train its large new Force and develop and try out its ships and gear. During these operations it was found that the cable and pipe would withstand all reasonable end-pulls, but that both would be severely kinked and damaged if allowed to hang vertically from the laying vessel, or if run back upon.

Full-scale trials were made with the Conun in the Thames in February 1944, and in Bournemouth Bay in April 1944, during which the technique for towing at speeds up to 7 knots was developed. The decision was also taken to moor the Conun at the beginning of her run and haul in the shore length of pipe by means of a warp pulled in by a plough traction engine, the farther end being laid parallel with the shore and subsequently pulled in.

It was the decision to lay Pluto lines to Cherbourg which had made necessary much larger supplies of Hais cable and Hamel pipe; and, in addition to increasing the British manufacture of Hais cable as much as possible, and starting production in the U. S. A., the decision was taken to duplicate the Tilbury facilities for welding and winding Hamel pipe.

I will now refer briefly to an American scheme which had been developed for laying cross-Channel lines by welding lengths of about 6 miles of 6-inch steel pipe on shore and towing them out to sea under Radar guidance in the hope that the ends might be connected below water by means of extremely ingenious "grasshopper" swivel joints. When it was seen how far Hais and Hamel had been developed and that no joints were required in the main lays, and realizing the difficulties divers would encounter while making joints in the tidal currents of the English Channel, General Eisenhower decided to abandon the American scheme in December 1943, and to give us all possible help in making Hais cable and in providing pumping equipment, etc.

I have reserved a reference to one aspect of the whole operation until now in order to emphasize its extreme importance. I refer to security and camouflage. Unlike many war secrets, Pluto could have been given away by a single phrase, for instance: "A petrol pipe like a hollow submarine cable across the

Channel", and pumping stations and both types of sea lines could easily have been attacked.

The pumping station construction was under the supervision of a camouflage officer; all plant which might be seen from the air was moved into position at night; and existing buildings such as bungalows, garages, ice cream factories, etc., were used as pump houses. Control photographs were taken at regular intervals by the R. A. F., and the complete success of all the precautions, often expensive and irksome, was proved by the lack of any known attempt by the enemy to interfere at any time.

All was ready to carry out Operation Pluto some weeks before D-Day, but it could not begin till 12th August 1944, because of the delay in capturing Cherbourg and in clearing sea mines. Two Hais cables and two Hamel pipes were laid during the next few weeks and difficulties were encountered which provided valuable experience for the future lays.

Petrol was pumped across to Cherbourg, but the Allied Armies' rapid advance along the French Coast made it necessary to concentrate all efforts on the Dungeness crossing, and the Cherbourg lines were shut down.

The Dungeness lines were run to a beach inside the outer harbor of Boulogne in order to save the time required to clear the heavily mined beach at Ambleteuse which had previously been chosen. This involved a longer run and a more difficult approach, but the first was laid in October 1944 and after the technique had been perfected for laying the main lengths of Hais cable over the stern and dropping the ends to the bottom, to be picked up at a suitable state of a later tide for connecting to the shore ends in the barges, further lines were laid and commissioned with certainty and without incident.

The method of pulling in the Hamel shore ends from the Conun, however, proved difficult and involved the loss of one Conun. A solution was found by winding turns of Hais cable at the beginning and end of each length of Hamel, followed by a special floating wire. The Conun could then be handled like the cable ship, laying each end on the bottom for the barges to pick up and connect to the shore ends in the same way as for a complete Hais. This technique enabled the first Hais-Hamel line to be commissioned in January 1945.

Force Pluto were responsbile for laying the line to above low water on each shore, and the R. E. and R. A. S. C. then connected the home end with steel pipe to the valves and filters provided on the pump delivery lines, and at the far end to a valve manifold.

Main and group control rooms, with telephone communication between themselves, the pump houses, and to the opposite terminal, were provided, and had diagrams on their walls on which the control officers could indicate by disks on hooks the direction of flow of oil, as well as the pumps and lines in use, etc., at any time.

Reference has been made earlier in the paper to the bursting disks fitted to the Hais cable joints to contain water under pressure in the cables while being laid and connected. When a line was ready for commissioning, a pump was started and the rate of rise of pressure was recorded. At first the rise was slow, and when a pressure of 400 lb. per sq. in. or so was reached, the first disk was broken and the pressure was seen to fall immediately, and then begin slowly to rise again. This was repeated at each disk until the arrival of liquid was at the farther end reported by direct telephone.

Each 3-inch line at Dungeness could deliver about 400 tons a day, or 120,000 gallons, and lines were laid quickly enough to keep ahead of the capacity to pump beyond Boulogne. Eventually eleven Hais and six Hais-Hamel lines were laid with a capacity of more than 4500 tons, or 1,350,000 gallons a day; and 1,000,000 gallons a day were pumped across for many weeks.

The valve manifold system on the beach at Boulogne, with a tank at beach level, provided facilities for test purposes, but the flow was usually taken direct through three lines of 6-inch Victaulic-jointed pipe to tanks of 1200 tons capacity on the cliff north of Boulogne. The petrol was then pumped to Calais, Ghent, Antwerp, Eindhoven, and across the Rhine through further 6-inch Victaulic lines.

A duplicate system of lines was being laid right up to "V. E."-Day to provide against a possible last desperate submarine attack, and pumping was continued until the end of July 1945, to enable all available tankers to be diverted to the Far East.

In the words of the Quartermaster-General, " . . . it saved a very large tanker tonnage which was badly needed in the East. Thus Pluto had a reaction which extended all over the world".

No Hais cable which had been satisfactorily laid and commissioned failed; and a recovery operation to clear the beaches has recently shown the cable to be in good and usable condition. There is every indication that, like submarine telephone cables on which it is modelled, it will have a long life. The Hamel pipes on the other hand, while more than fulfilling the original estimate of six weeks of useful life, did fail successively in 77, 52, 55, 112, 55, and 60 days. The loss of petrol between tanks at Dungeness and at Boulogne was, however, less than 1.1 per cent for the whole operation, in spite of these failures.

The Supreme Allied Commander, General Eisenhower, described Pluto in his report as "second in daring only to the artificial harbors projects", and wrote, "This provided our main supplies of fuel during the winter and spring campaigns".

I hope I have been able to give you some idea of the really remarkable co-operation and whole-hearted effort on the part of all the varied interests involved, which was necessary to meet the operational requirement and to enable Pluto to deliver 172,000,000 gallons to the Allied Armies of Liberation with certainty and at the daily rate required.

TORSIONAL VIBRATION STRESS IN SHAFTING.

In this article, which is an extract from a paper read at the spring meeting of the I. N. A. reprinted from "The British Motor Ship" for May, 1947, torsional vibration stresses in main propulsion shafting are discussed by S. F. Dorey, Chief Engineer Surveyor of Lloyd's.

Propelling Machinery. Limits of Vibration Stress for Continuous Operation.

Taking the fatigue strength of a 10-inch diameter solid forged crankshaft of 28-32 ton quality at ± 6000 lb. per sq. in., it is considered that the minimum factor of safety to allow for additional bending fatigue, imperfect forging methods, etc., should be about 1.5. A limiting stress of ± 3700 lb. per sq. in. has been finally adopted, giving a nominal factor of safety of $\frac{6000}{3700} = 1.62$.

From considerations of scale effect, increased forging difficulties, etc., the limiting stress for a very large shaft, say 20 inches in diameter, has been taken as ± 3000 lb. per sq. in. Scale effect for other shaft sizes is difficult to estimate and in the absence of reliable test figures has been assumed to vary linearly with diameter, leading to the following simple formula for limiting vibration stress under continuous operation:—

$$f_e = \pm(4400 - 70\,d) \text{ lb. per sq. in.} \qquad (1)$$

where d = shaft diameter in inches.

The above formula is taken as applicable also to screw shafts for the following reasons:—

(i) Liability to corrosion fatigue due to sea-water action from leaky rubber sealing rings, slack or cracked liners, defective oil glands.

(ii) Stress concentration at propeller keyways and at the top of the cone where a fillet is employed. The latter practice, incidentally, is one which the author considers highly undesirable.

(iii) It has been found that under unfavorable conditions the vibration stress required to fracture a screw shaft is of about the same magnitude as the lowest stress which will break a crankshaft.

Taking the fatigue strength of a 10-inch diameter intermediate shaft of 28-32 ton quality at ±8300 lb. per sq. in., a limiting stress of about ±6100 lb. per sq. in. has been adopted, giving a nominal factor of safety of 1.35. The reduced factor of safety as compared with that of 1.62 allowed for crankshafts is considered justified having regard to the negligible bending fatigue stresses in intermediate shafting, even under fairly severe conditions of malalignment.

It was considered that for very large shafts, say, of the order of 20 inches diameter, the vibration stress should not exceed ±5000 lb. per sq. in., and again assuming a linear relationship for other diameters, the following formula has been derived, viz:—

$$f_e = \pm(7320 - 116\,d) \text{ lb. per sq. in.} \qquad (2)$$

where d = shaft diameter in inches.

The recommended limits of vibration stress for thrust shafts for continuous operation have been taken as approximately equal to the mean of the values applicable to screw shafts and intermediate shafts, respectively, leading to the formula:—

$$f_e = \pm(6000 - 100\,d) \text{ lb. per sq. in.} \qquad (3)$$

where d = shaft diameter in inches.

Limits of Vibration Stress for Transient Criticals.

In most cases it is possible to arrange the shaft system so that the vibration stresses in the vicinity of the service speed are within the limits given above for continuous operation. However, this frequently involves the acceptance of important criticals exceeding these limits elsewhere in the speed range. In general, it is preferable to place such criticals well above service speed, as is sometimes possible for 1 node vibrations in aft end installations, such as tankers, by fitting oversize shafting.

In other cases, however, the critical will occur below normal service speed. By a transient critical is meant one which is too great for continuous operation and must be negotiated or passed through as rapidly as possible when maneuvering, in order to limit the number of cycles of high stress, imposed on the shafting.

Owing to the "spread" of the resonance curve, there will clearly be ranges of speeds above and below the critical which must be avoided for continuous running. These are sometimes referred to as "barred" speed ranges.

For transient criticals it is necessary to define limits for the following:—

(*a*) The maximum value of the transient vibration stress for a given position of the critical in the speed range.

(*b*) The corresponding "barred" speed range.

Clearly it is undesirable that a transient should be located too close to the service speed; hence limits have been chosen for the minimum interval between the transient and the service speed. At the lower end of the speed range a value of 0.27 times service Rpm. is taken as the minimum "tick-over" speed based on large slow-running machinery, such as opposed piston engines.

Transient Stress Limits.

For a 10-inch diameter crankshaft the maximum value of stress due to a transient critical occurring at "tick-over" speed, 0.27 times service Rpm., has been taken as about 1.5 times the nominal fatigue limit and the limiting stresses for transients occurring elsewhere in the speed range have been taken as decreasing proportionally down to f_e at 0.9 times service Rpm.

In order to preserve the allowance for size effect, the limiting transient vibration stress is expressed in terms of f_e, thus:—

$$f_t = \pm(3.07 - 2.3\,r) \times f_e \text{ lb. per sq. in.} \qquad (4)$$

where $r = \dfrac{\text{critical Rpm.}}{\text{service Rpm.}}$, having limiting values of 0.27 and 0.9.

f_e = limiting stress for continuous operation from formula (1).

Typical graphs showing values of f_c and f_t for shaft diameters of 4 inches and 18 inches are given in Fig. 1, the respective values being as follows:—

Diameter (ins.)	f_c (lb./in.²)	f_t (at $r = 0.27$) (lb./in.²)
4	±4,120	±10,000
18	±3,140	± 7,700

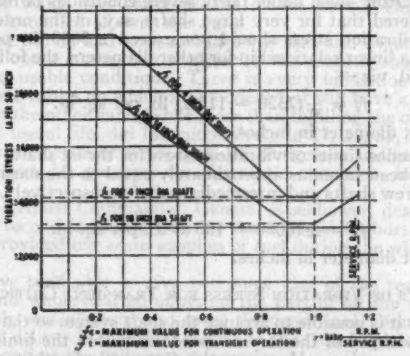


Fig. 1.—Torsional vibration stress. Crank- and screw shafts. Typical graphs giving values for 4-in. and 18-in. dia. shafts.

For intermediate shafts a similar basis has been adopted, i.e., f_t is expressed in terms of f_c, but in this case between limits for r of 0.27 and 0.75 to allow for the greater "spread" of transient criticals having higher limiting values than in the case of crankshafts and screw shafts.

For a 10-inch diameter shaft at $r = 0.27$ a limiting stress of ±12,300 lb. per sq. in. has been adopted, again approximately 1.5 times the nominal fatigue limit as in the case of crankshafts.

The corresponding formula for limiting transient vibration stress is:—

$$f_t = \pm(2.56 - 2.08\,r) \times f_c \text{ lb. per sq. in.} \qquad (5)$$

where r = ratio $\dfrac{\text{critical Rpm.}}{\text{service Rpm.}}$, having limiting values of 0.27 and 0.75.

f_c = limiting stress for continouus operation from formula (2).

Typical graphs showing values of f_c and f_t for shaft diameters of 4 inches and 18 inches are given in Fig. 2, the respective values being as follows:—

Diameter (ins.)	f_c (lb./in.²)	f_t (at $r = 0.27$) (lb./in.²)
4	±6,860	±13,700
18	±5,230	±10,450

The variation in the limits for transient stresses for thrust shafts has been taken on a similar basis, the formula being as follows:—

$$f_t = \pm(2.7 - 2.07\,r) \times f_c \text{ lb. per sq. in.} \qquad (6)$$

where f_c = limiting stress for continuous operation from formula (3), and r has limiting values of 0.27 and 0.82.

Typical graphs showing values of f_c and f_t for shaft diameters of 4 inches and 18 inches are given in Fig. 3, the respective values being as follows:—

Diameter (ins.)	f_c (lb./in.²)	f_t (at $r = 0.27$) (lb./in.²)
4	±5,600	±12,000
18	±4,200	± 9,000

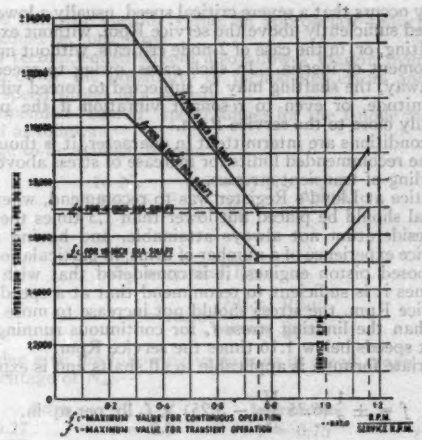


Fig. 2.—Typical graphs giving values for 4-in. and 18-in. dia. intermediate shafts.

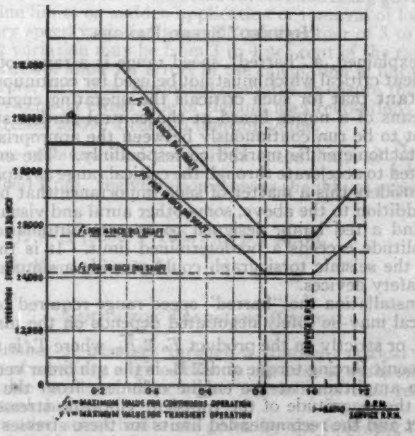


Fig. 3.—Typical graphs giving values for 4-in. and 18-in. dia. thrust shafts.

It frequently occurs that a severe critical speed, usually a lower order major, cannot be raised sufficiently above the service Rpm. without excessive stiffening of line shafting, or, in the case of 2-node criticals, without undue reduction in flywheel moment of inertia. In such cases, owing to speed surges when racing in a seaway, the shafting may be subjected to forced vibration of considerable magnitude, or even to resonant vibration if the position of the critical is unduly close to the service Rpm.

Since such conditions are intermittent in character, it is thought legitimate to deal with the recommended limits for increase of stress above service Rpm. under the heading of transient stresses.

Earlier practice at Lloyd's Register was to recommend, wherever possible, that the critical should be placed not lower than 1.3 times the service Rpm. This was a desideratum not always attainable, and having regard to the successful service experience of a number of firms, in particular one well-known builder of opposed piston engines, it is considered that with modern well-governed engines it is sufficient to recommend that at a speed equal to 1.16 times the service Rpm. the stress should not increase to more than 33⅓ per cent greater than the limiting stress f_o for continuous running, and proportionally less at speeds below 1.16 times the service Rpm.

The appropriate formula is applicable to all shafts and is expressed as

$$f = \pm \frac{1}{3} (6.25 \frac{N}{N_s} - 3.25) \times f_o \text{ lb. per sq. in.} \qquad (7)$$

where

f = maximum value of vibration stress at revolutions N, above service Rpm., N_s.

f_o = appropriate limiting stress for continuous operation from formulae (1), (2) or (3).

It will be noted this formula takes account also of flank stresses and whilst not specifically limiting the position of a critical, has the effect that strong criticals are required to be placed farther above service speed than weaker ones. Where it can be demonstrated that governing is specially close, some reduction might be considered in the above recommended limits.

"Barred" Speed Ranges.

As already explained, a "barred" speed range is a range of revolutions in way of a transient critical which must not be used for continuous running. It is most important that for such criticals the operating engineers should be warned by means of a notice board at the control station stating that the engines are not to be run continuously between the appropriate speed limits and that the tachometer be marked correspondingly. The engineers should also be instructed to accelerate through the critical range as rapidly as possible. The author considers this a matter of such importance that he would like to see fitted, in addition to the above, some other aural and visual warning, such as a buzzer and a red lamp, designed to operate automatically when the vibration amplitude exceeds a predetermined limit. It is thought that an adaptation of the seismic torsiograph could provide a simple means of controlling such safety devices.

In a given installation the "barred" speed range required to ensure that a transient critical may be safely negotiated depends on the magnitude of the forcing torque, or strictly on the product $T_n \Sigma \overline{B}_n$, where T_n is the value of the nth order harmonic forcing torque and $\Sigma \overline{B}_n$ is the nth order vector summation of the relative amplitudes at the engine cylinders from the normal elastic curve. Since the magnitude of the transient vibration stresses also depends on this product and the recommended limits for these stresses increase as the position of the critical falls in the speed range, it is considered reasonable that the extent of the "barred" speed range, expressed as a fraction of the critical speed, should also increase. Where, however, the severity of a transient

critical is less than the limiting value, an appropriate reduction in "barred" speed range could be considered.

Any limits recommended should also allow a margin over and above the minimum, to cover the possibility of tachometer errors and, in particular, unavoidable speed surges in a seaway.

The recommended "barred" speed ranges for the 1-node mode of vibration are expressed as follows:—

Speed range to be avoided is from

$$\frac{16\ N_e}{(18-r)} \text{ to } \frac{(18-r)\ N_e}{16} \text{ inclusive} \qquad (8)$$

where

N_e = critical Rpm.

r = ratio $\frac{\text{critical Rpm.}}{\text{service Rpm.}}$

The formulae give the following variations in speed range, expressed as an average percentage of N_e:—

Screw		Intermediate	Thrust
$r = 0.27$	$r = 0.9$	$r = 0.75$	$r = 0.82$
± 10.3	± 6.7	± 7.5	± 7.1

For 2-node criticals a constant value of ± 5 per cent is considered appropriate, having regard to the lower values of $T_e \Sigma \bar{E}_e$, for these criticals, which are sharper tuned, i.e., have less "spread" on account of the lower values of the harmonic torques for the higher orders applicable to 2-node vibration.

Auxiliary Machinery Stress Limits for Continuous Operation.

An auxiliary generating set is, for all practical purposes, a constant-speed machine, the fluctuations from the rated revolutions being governor controlled within fairly fine limits on sudden application or removal of load. The maximum temporary speed variation is usually of the order of 5 to 7 per cent, and the permanent variation may be from 3 to 4 per cent of the rated revolutions. However, in order to allow for errors in the calculation of natural frequencies, tachometer errors, etc., it is considered desirable to ensure that a speed range of about 10 per cent on either side of the full load revolutions should be clear of vibration stresses above a certain limiting value.

It is therefore recommended that within speed limits of $\frac{N_e}{1.1}$ and $1.1\ N_e$, N_e being the full load Rpm., the vibration stresses should not exceed the following values, viz.:—

$$f_e = \pm(5000 - 100d) \text{ lb. per sq. in.} \qquad (9)$$

where d = shaft diameter in inches.

Limits for Transient Criticals.

Since the number of passages through a transient critical is considerably less in the case of auxiliary engines than in the case of main engines, it is considered reasonable to adopt rather higher stress limits than recommended for main engine crankshafts.

The recommended stress limits are given by the following formula, viz.:—

$$f_t = \pm(15{,}000 - 250\,d) \text{ lb. per sq. in.} \qquad (10)$$

where d = shaft diameter in inches.

The stress values are slightly greater than those applicable to transients in intermediate propelling shafts for a value of $r = 0.27$.

For a 6-inch diameter shaft the above formula gives a stress of $\pm 13{,}500$ lb. per sq. in. as compared with $\pm 13{,}240$ lb. per sq. in. for a 6-inch diameter intermediate shaft for $r = 0.27$.

It may be of interest to quote here the results of an important test recently carried out in co-operation with Messrs. W. H. Allen, Sons and Co., Ltd., of Bedford (to whose courtesy the Society is indebted for the facilities granted). One of this firm's eight-cylinder engines, having a solid forged crankshaft 5½ inches (140 mm.) in diameter, with a ratio of shaft diameter to fillet radius of 17.5, was run continuously on the 4th order major transient critical until failure occurred. The measured stress at the critical was $\pm 15{,}000$ lb. per sq. in. in the crankshaft, which was of 30-35-ton quality steel. After 75,000 cycles a definite change of natural frequency was observed and this number of reversals was taken to indicate initiation of failure. After a further 75,000 cycles, the engine was opened up and examination of the shaft revealed cracks in the fillet radii. The test was then continued for a further 150,000 reversals before bearing failure due to shaft deformation terminated the test, the total number of cycles being 300,000.

On the conservative basis of a life of 75,000 cycles and assuming one start and stop per week, with an average of, say, 40 cycles of high stress per start and stop (this was actually measured at 35 cycles per start and stop), then the estimated life of crankshaft is given by

$$\frac{75{,}000}{40 \times 52} = 36 \text{ years}$$

It would thus appear that the recommended limits for transient vibration stresses may lie on the safe side, where care is taken in design and attention paid to surface finish and easing of stress raisers.

Vibration Dampers.

For both main and auxiliary engines it is recommended that dampers be fitted where transient vibration stresses would otherwise exceed the recommended limits, and that torsiograph records be taken to verify the correct adjustment of the damper.

The author would emphasize this latter point, since in a number of cases it has been found by actual measurement that the damper was entirely ineffective, and, in some instances, of inadequate size.

Having regard to the difficulty of maintaining the correct adjustment of dampers in service, the author is of the opinion that, in general, they should be fitted to deal with transient stresses rather than for continuous operation in the vicinity of the service speed.

In some cases it has been found that tachometers have shown errors up to 5 per cent. Where transient criticals require to be avoided, it is therefore recommended that the tachometer should be checked against the counter readings to verify that the tachometer reads correct within ± 2 per cent of the true speed in way of the restricted range of revolutions.

Propellers.

In the smaller class of vessel, such as coasters, colliers, etc., it is common practice to fit a trial propeller of cast iron, which is later replaced by one of manganese bronze after an analysis of the trial or initial service performance has been made. From the point of view of the shipowner, this is obviously an economy, but, unfortunately, in some cases the moment of inertia of the bronze propeller may differ appreciably from that of the trial cast-iron one. Occasionally this has had the effect of bringing a major critical uncomfortably close to the service Rpm.

It is therefore recommended that designers should bear this in mind when making the critical speed calculations.

The author would like to suggest also that where the calculated position of a critical is only just sufficiently far removed from the service Rpm., it is good practice to "swing" the propeller in the shop as an experimental check on its calculated moment of inertia.

Most modern oil engines are fitted with governors capable of controlling speed surges to within 5 per cent when racing. Some confusion recently arose when it was stated that the governors in a certain trawler installation would be set to cut off fuel at certain specified revolutions. It is hardly necessary to point out that what really matters is not the speed at which the governor cuts off fuel, but rather, the maximum speed which the engines can reach under governor control when the load is suddenly thrown off, as when the propeller lifts during racing.

Crankshaft Fillet Radii and Oil Hole Diameters.

In all cases fillet radii and oil hole diameters should be maintained as large as practicable.

For crankshafts of normal 28-32-ton quality recommended limits are as follows:—

Ratio $\frac{\text{crankshaft diameter}}{\text{fillet radius}}$ should not exceed a value of 20.

Ratio $\frac{\text{crankshaft diameter}}{\text{oil hole diameter}}$ should not exceed a value of 20.

Where higher tensile or alloy steels are employed, these ratios should preferably be further reduced.

The endeavor of Lloyd's Register of Shipping is to ensure that every marine installation is reasonably secure from danger of shaft failures, whether resulting from torsional vibration or from other causes, and this, the author confidently believes, is an aim which must be shared by those builders and owners who take the long view, namely, to secure the continued competitive position of the heavy-oil engine in the marine field. It is for this reason that the author wishes to obtain from industry the fullest possible criticism and discussion on the proposed basis of torsional vibration stress limits now placed before this Institution.

AN UNUSUAL BOILER MISHAP.

This article by C. E. Hall is reprinted from the Transactions of the Inst. of Marine Engineers for March, 1947. It describes a boiler fire in which the tube bank burned out almost completely. The German cruiser *Prinz Eugen* had a similar fire in one of her La Mont boilers.

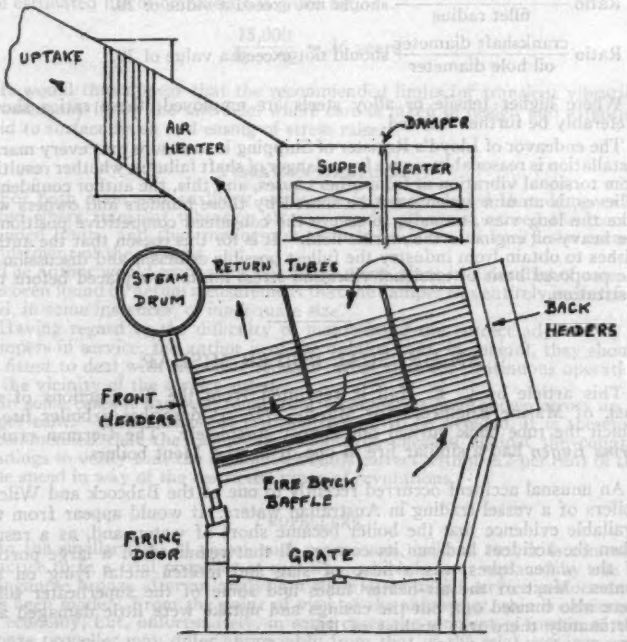
An unusual accident occurred recently to one of the Babcock and Wilcox boilers of a vessel trading in Australian waters. It would appear from the available evidence that the boiler became short of water, and, as a result, when the accident had run its course, all that remained of a large portion of the water tubes was a heap of slag and molten metal lying on the grates. Most of the air-heater tubes and some of the superheater tubes were also burned out, but the casings and uptake were little damaged and fortunately there was no loss of life.

The vessel had just been moored at the time of the mishap, and the boiler which subsequently burned out had been retained under natural draught to work auxiliaries. The first intimation that things were going wrong was a dimming of the electric lights, whereupon the fireman on watch shook up the fires and added more coal. Soon afterwards one of

the engineers tested the water gauges and, as he was not satisfied, called the chief engineer, who decided to shut down the boiler.

To cool the fires as quickly as possible, a hose was directed on to the coal on the grates, and when the fires were extinguished the only noticeable sign of damage was a trickle of water running down from the tubes. Almost immediately after this, an explosion occurred inside the casing and flames shot out of the furnace doors across the stokehold. These lasted for several minutes and then a crash was heard, whereupon the rush of flame into the stokehold ceased and appeared to reverse and pass upwards to the funnel. This was accompanied by a roaring noise from inside the casing through the chinks of which glimpses of fierce flames could be seen. It was thought that this was due to accumulations of soot burning and it was expected that the flames would soon subside.

After about half an hour however, the combustion appeared to have become more intense and molten metal was observed dropping on the grates. As the uptakes were becoming red hot a hose was turned on them and as this had little effect, the hose was directed through a door in the uptake. The fire then gradually subsided and eventually died down altogether, although the interior of the boiler remained hot for a considerable period.



Arrangement of boiler.

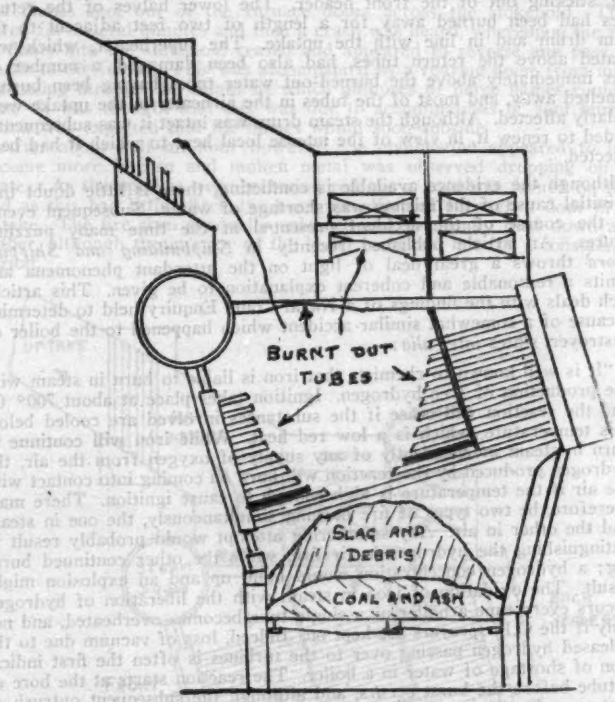
When examined externally after the mishap, the boiler appeared to have suffered little damage. The outer casings were intact although the uptake was warped. It was not possible to see into the furnaces as the furnace

doors were blocked by a mass of slag and molten metal. A side door in the upper part of the boiler casing allowed an examination to be made of the interior and from this position it could be seen that most of the water-tubes had disappeared. All that remained were the lower row, that portion of the tubes enclosed in the back gas passage, and stumps, some six inches long, sticking out of the front header. The lower halves of the return tubes had been burned away for a length of two feet adjacent to the steam drum and in line with the uptake. The superheater, which was situated above the return tubes, had also been damaged, a number of tubes immediately above the burned-out water tubes having been burned or melted away, and most of the tubes in the airheater in the uptake were similarly affected. Although the steam drum was intact it was subsequently decided to renew it, in view of the intense local heat to which it had been subjected.

Although the evidence available is conflicting, there is little doubt that the initial cause of the mishap was shortage of water. Subsequent events and the course of the accident presented at the time many puzzling features. An article published recently in *Shipbuilding and Shipping Record* throws a great deal of light on the attendant phenomena and permits a reasonable and coherent explanation to be given. This article, which deals with the findings of a Naval Court Enquiry held to determine the cause of a somewhat similar accident which happened to the boiler of a destroyer, states *inter alia*:—

"It is well known to chemists that iron is liable to burn in steam with the production of free hydrogen. Ignition takes place at about 700° C., and the reaction will cease if the substances involved are cooled below this temperature, which is a low red heat. While iron will continue to burn in steam independently of any supply of oxygen from the air, the hydrogen produced by the reaction will burn on coming into contact with the air if the temperature is high enough to cause ignition. There may, therefore, be two types of fire burning simultaneously, the one in steam and the other in air. Any smothering attempt would probably result in extinguishing the hydrogen-in-air fire, while the other continued burning; a hydrogen concentration would build up and an explosion might result. The oxidation of iron in steam with the liberation of hydrogen occurs every time a boiler or a single tube becomes overheated, and not only if the O.F. sprayers are kept on. Indeed, loss of vacuum due to the released hydrogen passing over to the turbines is often the first indication of shortage of water in a boiler. The reaction starts at the bore of a tube before the burst occurs, and although the subsequent outrush of water cools the overheated tube below the ignition temperature and washes away the scale in the vicinity of the burst, black scale will always be found in the bore a short distance away on either side of the burst. If the source of heat is not removed and the supply of feed water is not maintained, the reaction will start up again. This reaction, which is the basis of Lane's process of hydrogen production, can be expressed in the formula: $3FE-4H_2O=Fe_3O_4-4H_2$, (Fe_3O_4) being the black oxide of iron sometimes known as ferrosic oxide. The reaction is exothermic, *i.e.* it produces heat and is self-sustained until such time as: (*a*) the temperature is reduced below 700° C., or (*b*) all the iron is oxidized, or (*c*) the steam supply is reduced. By this reaction, 168 lb. of iron, treated with the steam produced by 72 lb. of water, will liberate 8 lb. of hydrogen. At atmospheric temperature 8 lb. of hydrogen would fill a balloon 14 ft. in diameter. Steam can be dissociated into hydrogen and oxygen by heat alone, but only 0.0007 per cent is dissociated at 1124° C. (2050° F.) and 11 per cent at 2656° C. (over 4000° F.) so it is clear that thermal dissociation was not the cause of the above fire".

With these data it is possible to give a fairly satisfactory account of the phenomena observed in the boiler now under consideration, bearing in mind the fact that most of the events took place inside the boiler casing and the little that was visible was seen by one observer under very disturbing conditions.



After the mishap

Those who were present when the accident occurred were agreed that there had been no signs of escaping steam immediately before the accident, such as would be made by a tube bursting and causing a sudden lowering of the water level. The first relevant observation was the dimming of lights which, as suggested in the above report, may have been due to hydrogen passing over from overheated tubes and causing the dynamo to slow down. If that be the case, then the shortage of water must have been existent for some time before the fires were finally extinguished. The fact that the lower row of tubes remained intact seems to show that some water still remained in them at the end of the accident. How much water was evaporated during the course of the accident is conjectural. Taking into account all the relevant factors both before and after the accident, it may be taken as a reasonable assumption that the tubes in the lower half of the tube nest contained water while the upper ones had been dry for a period and had become red hot by the time the fires were quenched.

From the article cited above, it is clear that under such circumstances the steam inside the boiler would attack the inside surfaces of the tubes, causing an exothermic reaction accompanied by a rise in temperature. At the same time the outside of some of the tubes would be subject to a similar attack, where conditions were favorable, by the steam rising from the quenching of the fires.

The proviso "where conditions were favorable" raises an interesting point. In some experiments carried out after the accident it was observed that steam at 212° F. would cool red-hot iron, and it was necessary to heat strongly the steam to bring about an apparent reaction. While it is difficult to give a figure for the temperature of the steam evolved during the quenching of the fires, it is evident that this steam would be highly heated in rising through the red hot tubes in the back gas passage and in passing through the tubes in the middle gas passage. The heating of the steam would be accompanied by a corresponding cooling of the tubes in the back passage, and may account for the fact that these tubes remained intact. It therefore seems probable that the destruction of the water tubes started somewhere in the middle of the front gas passages, where conditions would be favorable to external as well as internal attack. Eventual rupture of one or more tubes would be caused by thinning of the tube walls and/or weakening due to overheating, causing the tubes to fail under even a slight internal pressure. The bursting of a tube or of several tubes in the center of a red-hot tube bundle, followed by a release of pent up hydrogen from inside the tube, makes the final destruction easy to understand.

In the conditions prevailing at the time there appear to be three forms of reaction which could have been taking place, individually in some places and simultaneously in others. These reactions would be (*1*) the burning of the released hydrogen in air, (*2*) the reaction between the iron and steam, and (*3*) the burning of iron in air.

In view of the large amount of heat released by the hydrogen plus oxygen reaction, it is likely that this was the cause of most of the subsequent damage. It is probable that the hydrogen which escaped when the tubes first failed, mixed with the air outside the tubes and caused the initial explosion which was observed. Such water as remained in the lower tubes would be rapidly evaporated and, passing through the remains of the red-hot tubes, would maintain the iron-steam reaction and release more hydrogen. There can be little doubt that it was this hydrogen generated from the escaping steam which gave rise to the fierce flames which shot out from the furnace doors. No doubt similar flames were passing upwards to the funnel but these would not be noticed from the stokehold floor.

The peculiarities of the iron-steam reaction have been made clear in the *Shipbuilding and Shipping Record* report quoted above, and the conditions for this reaction would become more favorable as long as the hydrogen-air reaction continued. Once the tubes had given way and started to melt, fresh red-hot surfaces would be exposed to the action of steam, increasing the fierceness of the reaction and thereby causing a further temperature rise.

In the circumstances prevailing, the possibility of the iron of the tubes burning in air must not be overlooked. The heat evolved is not very great but the heat lost would be small.

Normally, one does not consider that iron is combustible. Of course it can be burned in the abnormal conditions obtained with an oxy-acetylene torch and, as is equally well known, an overheated rivet will burn in air, though the action soon stops because the heat evolved in such an iron-air

reaction does not equal the radiation losses and also the reaction is quickly smothered by the formation of scale.

In the center of the nest of tubes there would be little loss of heat through radiation. The area of combustion was surrounded by red-hot tubes, the incoming air was highly heated by a rain of molten metal, and the melting tubes would be continuously exposing fresh molten-metal surfaces. All these factors indicate that part of the damage was probably caused by the iron-oxygen reaction.

These three actions would account for all the damage done to the various tubes in the boiler. The water tubes apparently failed first through the initial effect of the iron-steam reaction and were further burned and melted by the hydrogen-air reaction as well as by the iron-air reaction. There would be little steam in the superheater tubes, and as the damaged superheater tubes were confined to the bundle immediately over the place where the first tubes failed, it is probable that they were melted by the initial hydrogen-air flames, possibly accompained by some burning of iron in air. This would also account for the melting and partial burning of the air-heater tubes. Just how much damage was caused by the burning and how much by melting, it is impossible to say, and little attempt has here been made to distinguish the two effects.

While this conflagration was proceeding, a crash was heard and the flames ceased to issue from the fire doors and appeared to reverse and pass upwards towards the funnel. It is thought that this crash may have been caused by the superheater damper falling and the fire brick baffle over the front part of the lower tubes collapsing.

It seems likely that the front vertical baffle failed about this time, providing a clear passage to the uptakes and, for a period, the fire apparently increased in intensity under the influence of the various factors already discussed.

But the water remaining inside the tubes was steadily diminishing and the production of steam and hydrogen decreasing. When eventually a hose was directed inside the uptake, the stream of cold water, having a clear run down to the center of the fire, was sufficient to reduce the temperature below that necessary to maintain combustion.

In view of the extraordinary nature of this mishap, many hypotheses were examined before deciding that the process outlined above offered the most feasible explanation, conformable with observed events and with the final state of the boiler. The reactions involved are far from everyday processes; the necessary conditions are critical and depend on a nice balance of heat evolution and radiation. There are still points which are far from clear. It is a matter for wonder, for example, that the sides of the casing were not blown out in the first place, but when all the factors are considered it can only be said that it did not happen that way.

It is probable that data gathered from other mishaps of a like kind which have not yet been published, will affirm or refute some of the processes here set forth and shed further light on the phenomena which occurred.

BALL AND ROLLER BEARING SEALS.

In this article by E. P. Stahl of Garlock Packing Co., and reprinted from "Product Engineering" for April 1947, designs of common types of oil and grease seals for excluding foreign material from ball and roller bearings are shown. Proper application of different types of seals are discussed. Typical seal arrangements for industrial and automotive equipment are described.

Oil and grease seals in ball, roller and needle bearings must serve not only as lubricant retainers but also as excluders of foreign material. Properly designed bearing seals insure maximum protection of the bearing from the action of the grit, sand, mud and water that are prime causes of early bearing failure. Some of the simplest types of oil and grease seals designed to exclude foreign materials are shown in Fig. 1. Each type has a distinct field of application.

The "felt seal" or "combination felt type seal" is one of the most commonly used combination seals. It has a felt washer on the exposed face of the seal mounted in combination with a spring-loaded composition or leather sealing member. This seal has been used successfuly in automobile, truck, tractor and trailer wheels and automobile transmissions.

The "dual type combination seal" is used both for lubricant retention and dirt exclusion in steel mill and general industrial equipment exposed to excessive quantities of foreign matter. This seal construction incorporates an auxiliary sealing member or wiper ring that acts in conjunction with the primary spring-loaded sealing member.

In the "double type" or "double flange opposed type" seal, the sealing members are mounted in a single metal case in reverse tandem. One member retains the lubricant in the bearing and the other excludes foreign matter. To reduce seal friction and wear, lubrication between the sealing members is provided by drilling holes centrally in the outer periphery of the metal seal case. This type of seal has been used extensively in steel mill and general industrial equipment. Where space is available, two single seals mounted in reverse tandem with metal spacer buttons between them are now used more generally than the "double flange opposed type."

In the "special reversed element seal," in which the sealing member is mounted in the reverse position in the metal seal case, a small clearance between the metal case and the shaft diameter is used as an extra safety measure. These seals are used in applications such as ash conveyors in stokers and center propeller shaft steadying bearings on trucks where considerable dirt and abrasive are present.

The "external" oil seal or grease retainer is usually fitted on the shaft collar and operates in the ground or machined bore of the housing, just the reverse of the conventional shaft type seal. It is used in applications where a press fit on the outside diameter is undesirable or impossible, as in designs in which the axle spindle or shaft is stationary and the housing revolves and removability of the latter over the bearing is necessary. This seal has been used in aircraft landing wheels, trailer wheels and tractor sprockets and, in small sizes, for certain reciprocating applications.

Unit or mechanical oil or grease seals with composition or leather sealing members are delicate precision units developed primarily for fluid sealing on highly finished shafts. The primary sealing member, spring loaded, should be protected by a felt, leather or composition wiper as an auxiliary unit whenever harsh abrasives are present. A metal baffle plate or metal slinger may be used as a supplementary protective member when unusual conditions are encountered.

Space limitations and considerations of design sometimes prevent the use of special seals or special wide seal combinations. In applications of this kind a single conventional type seal mounted with the sealing lip pointed outward toward the atmosphere is used. Thus mounted, the seal helps keep out dirt and water. The slight escape of lubricant past the seal keeps foreign matter out of the bearing. The lubricant also lubricates

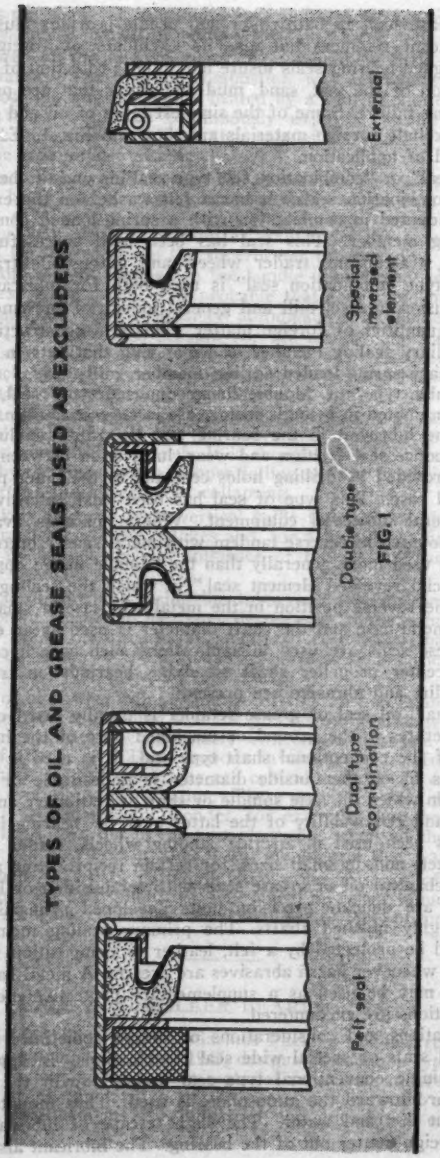

TYPES OF OIL AND GREASE SEALS USED AS EXCLUDERS
Felt seal
Dual type combination
Double type
Special inversed element
External

FIG. 1

the seal and reduces seal and shaft wear. Fig. 2 shows a typical application of such an arrangement in a screw type conveyor of a domestic stoker.

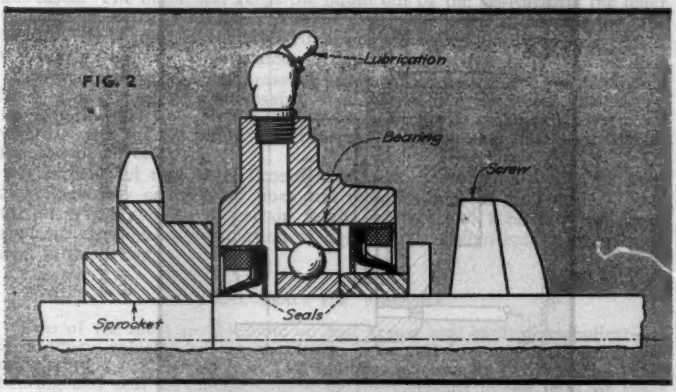


Fig. 2—Installation of conventional seals in screw conveyor of coal stoker.

Steel Mill and Industrial Applications.

In steel mill roll neck bearings, oil contamination and oil leakage through imperfect oil seals must be prevented. Oil contamination by certain roll coolants has been known to dilute the oil and cause bearing failure. It is sometimes necessary to renew whole systems of bearing oil because of oil dilution.

A satisfactory oil seal design for roll neck bearings using both composition and leather oil seals is shown in Fig. 3. Two mechanical unit oil seals are mounted in tandem with the sealing lips pointed outward to exclude water, dirt and mill scale. Spacer buttons welded to the seal cases are used between the oil seals so that lubricant can be introduced between them. A slight amount of the lubricant leaks out and lubricates the seal and at the same time helps to keep out foreign matter.

Submerged bearings for electrotinning lines in steel mills have always been more or less of a "headache" to design engineers and mill operators. Roll slippage, scratched sheets, scored journals and bearings, and high maintenance costs for journal bearing repairs and roll grinding were always encountered with plain sleeve type bearings of the water lubricated type.

In plain sleeve type bearings on Bonderizing rolls, abrasive sludge scored both the journals and the bearings. This sludge also built up on the roll body to form a hard, smooth coating that reduced the frictional grip between the roll and the sheet and resulted in long shut down periods for journal and roll repairs.

To overcome these difficulties the "submerged counter pressure bearing" shown in Fig. 4 was designed. Two needle type roller bearings mounted in a self-aligning bearing housing or sleeve carry the radial load of the Bonderizing rolls. Two mechanical unit oil seals mounted in reverse tandem are applied to this bearing housing at the inner end adjacent to the roll head. The inner seal lip faces away from the bearing and the outer seal lip faces toward the bearing to form an air pocket. An air pipe admits air at a pressure higher than the pressure of the liquid in the Bonderizing

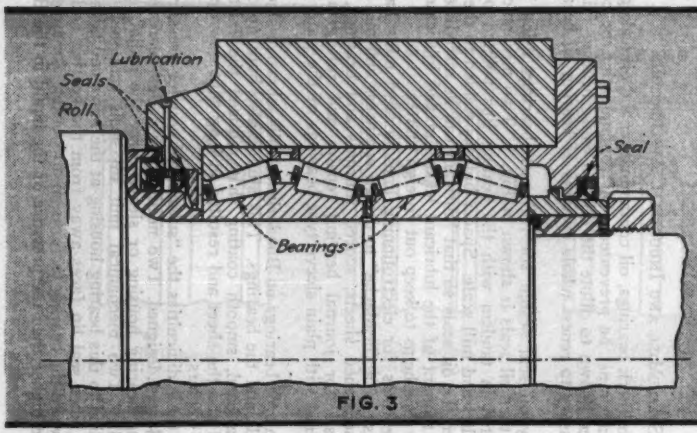


FIG. 3

Fig. 3—Tandem mounting of oil seals for steel mill roll neck bearings.

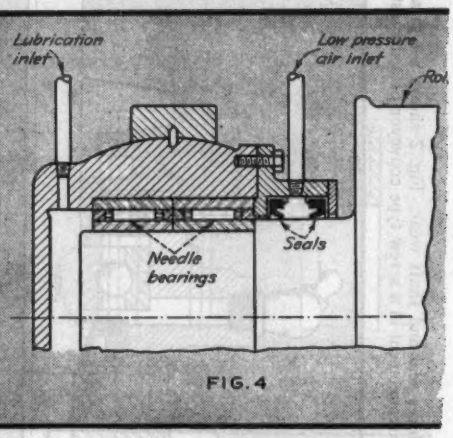


FIG. 4

Fig. 4—Counter pressure seals for bearings of Bonderizing rolls.

tank. This air pressure, of 5 to 25 lb. per sq. in., causes the lips of the synthetic rubber oil seals to hug the shaft and spread as the pressure is increased. The outer seal keeps the liquid out of the bearing and the inner seal acts as extra protection if the air pressure should fail or the air pipe break. The bearings are lubricated through the pipe at the outside end of the bearing housing as indicated.

The "submerged counter pressure bearings" on the rolls of electrotinning lines have proven very successful. Full line speeds have been maintained for a period of six months or more with practically no maintenance costs whereas with the old sleeve type bearings it became unprofitable to run at the end of three days.

It is believed that these "counter pressure bearings" can also be used advantageously on cleaning lines and pickling lines in steel mills. Their use may be extended to certain textile, marine and steel mill roll neck applications where the exclusion of liquids and foreign matter from bearings is important.

Oil Seals for Vehicles.

Two of the most troublesome oil and grease seal exclusion applications in military vehicles are in the track links and in the bogey wheels in crawler type vehicles such as gun mounts, tractors and half-tracks. The bearing and seal arrangement used by the Germans in the links of their ¾ ton tractors and half-tracks is shown in Fig. 5. The superior performance of these tractors and half-tracks is attributed to the links, which are equipped with needle roller bearings. The needle bearings in these units are sealed with Perbunan synthetic rubber seals mounted with the sealing lips pointed outward to exclude grit, sand, mud and water. According to captured German crews these links were never lubricated in the field since the factory lubricant was adequate.

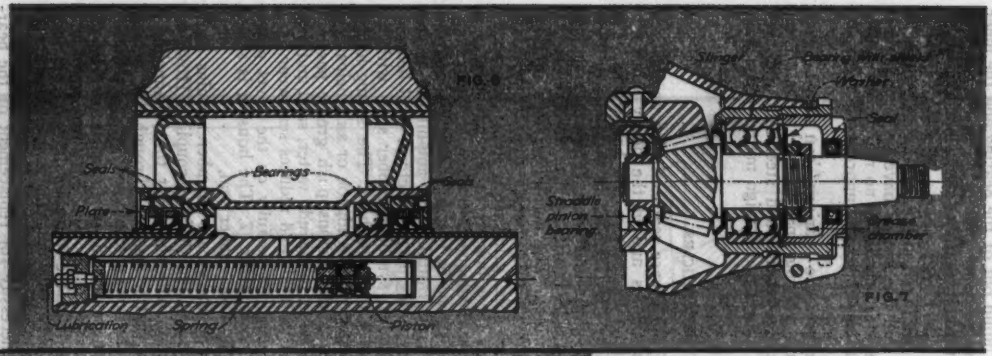
The bearing and seal arrangement shown in Fig. 6, which was used for the bogey wheel of the famous "Water Buffalo"—an amphibious tractor used by the U. S. Army and Navy, was highly satisfactory. Ball bearings are mounted in combination with tandem synthetic rubber seals with the sealing lips pointed outward toward the atmosphere. The spring-loaded piston with leather washers in the bogey wheel gudgeon or axle helps keep out water. When packed with grease, the pistons push grease past the seals. The escape of grease not only helps to exclude water, sand and mud from the bearings but also lubricates the seals and reduces seal wear.

A bearing and seal arrangement used successfully on the bevel pinion shaft on automotive rear axles is shown in Fig. 7. The housing is sealed with a spring-loaded, leather or composition rubber seal.

The greater portion of the grease in the grease chamber remains static and is not agitated or churned by the rotating shaft or coupling. However, a small part of the grease will rotate with the shaft thus channeling a labyrinthian passage. Any foreign matter passing through the spring-loaded housing seal is trapped in the grease contained in this chamber.

A cross-sectional view of a bearing and seal arrangement used on the rotors of a magnetic separator is shown in Fig. 8. Two single unit mechanical seals, mounted on each side of the self-aligning roller bearings, have provision for injecting grease between them. These machines handle abrasive sand and are greased only weekly. Some of them have been in operation eleven years without any necessity for replacement of the bearings or seals.

A special composition seal applied to a self-aligning roller bearing is shown in Fig. 9. This seal, which takes care of maximum misalignment, is designed to contact a spherical shaft collar at a tangent and has been very effective as a dirt and water "excluder."



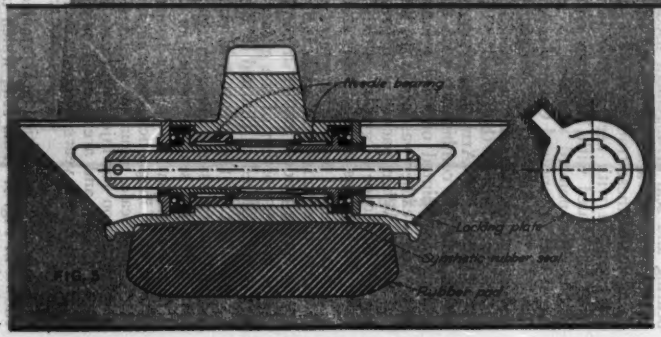


Fig. 5—Seals for track link of German ¾ ton tractors and half-tracks.

Fig. 6—Bearing and seal arrangement for Bogey wheel of "Water Buffalo" amphibious tractor.

Fig. 7—Bearing and seal arrangement of the bevel pinion shaft of automotive rear axles.

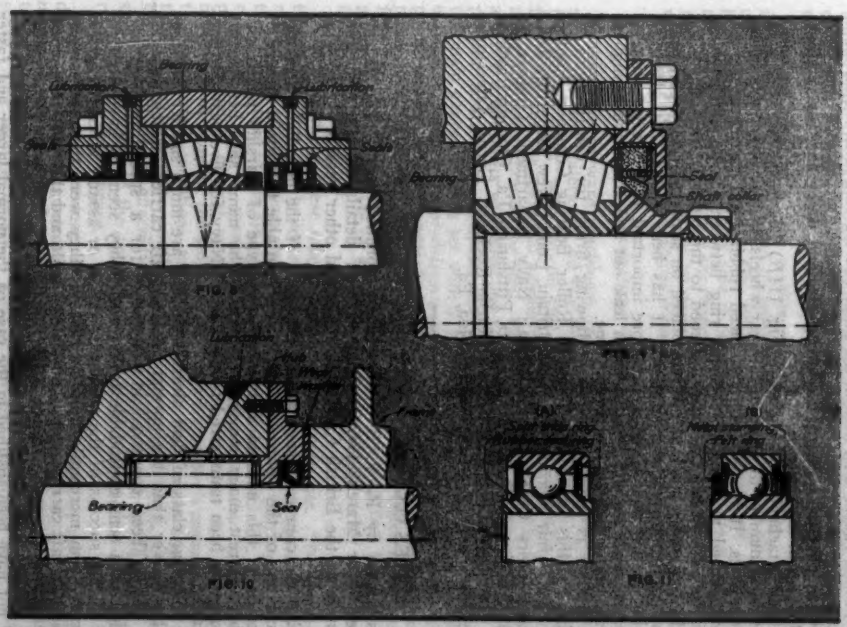


Fig. 8—Bearing and seal arrangement of rotor shaft of magnetic separator. Fig. 9—Special seal arrangement for self-aligning roller bearings. Fig. 10—Bearing and seal arrangement of guide-roll of road roller. Fig. 11—Detachable and non-detachable seals for sealed ball bearings.

A sectional view of a bearing and seal arrangement in the guide roll of a road roller is shown in Fig. 10. The seal is mounted with the sealing lip pointed outward to exclude dirt and water. Any slight escape of grease under the heel of the seal not only helps to exclude foreign matter from the bearing but also lubricates the wear washer.

A composition rubber detachable seal for a sealed ball bearing used extensively in aircraft controls is shown in Fig. 11(*A*). This seal consists of a synthetic rubber disk held in the outer bearing ring with a split metal retaining ring. This seal, which operates on the "air cleaner" principle, has proven effective both for excluding dirt and water and for retaining the lubricant in the bearing.

The non-detachable seal shown in Fig. 11(*B*) has been used with good success for a number of years in the rear wheels of several popular makes of automobiles. It consists of a felt faced rubbing seal carried in a metal slinger attached to the inner bearing ring between the annular opening formed by two sheet metal parts fastened to the outer bearing ring.

In the past years much stress was placed on seals for sealed ball bearings as lubricant retainers. Recently it has been found that often the exclusion of dirt and water is far more important than the retention of grease. A small amount of grease can leak out and still leave enough to take care of the bearing for a long time.

The author is indebted to the following firms for contributions of material for this paper: The Timken Roller Bearing Company, Canton, Ohio; The Fafnir Bearing Co., New Britain, Conn.; New Departure Div., General Motors Corp., Bristol, Conn.; Navy Department (Bureau of Ships), Washington, D. C.; W. T. Hurtt, Pittsburgh, Pa., Chicago Rawhide Mfg. Company, Chicago, Illinois; Marlin-Rockwell Corp., Jamestown, N. Y.; SKF Industries, Inc., Philadelphia, Pa.

NOTES ON LIGHT FLEET AIRCRAFT CARRIERS.

This article by R. V. B. Blackman is reprinted from "The Marine Engineer" for March, 1947, and gives brief details of the propelling machinery of the British Carrier *Colossus* and other vessels.

During the course of several visits to four of the new light fleet aircraft carriers of the *Colossus* class I have been able to form a picture of the design of this unusual and unorthodox type of warship which will probably interest many marine engineers. The name ship herself, the *Colossus*, has been transferred to the French Navy for five years, it may be recalled.

The *Colossus* class of fleet aircraft carriers are much smaller and considerably slower than the normal fleet aircraft carriers which preceded them, having been conceived in emergency for a special purpose and regarded as a semi-expendable task force. They seem to have been inspired by the success of the makeshift escort or assault aircraft carriers converted from mercantile hulls in 1941-42; they were the smallest and cheapest aircraft carriers it was possible to build and the most economical to operate, being capable of the triple functions of fleet, assault and escort work. Risks could be taken with these light fleet aircraft carriers which would not have been justified in the case of the normal larger and faster fleet carriers.

In their design merchant ship practice was followed where simplicity in construction and the use of available material and labor resources would facilitate rapid production.

The propelling machinery is of straight forward type and contained in only two spaces, instead of upwards of four compartments, as is the case in other warships of comparable dimensions.

The two machinery spaces are designed on the unit system, each being self-contained, the four Admiralty three-drum boilers and two sets of Parsons type geared turbines being arranged *en echelon*, with a pair of boilers and a set of turbines, with all its auxiliaries, in each space.

The open-front 10,000 to 12,000 Hp. boilers, each with five oil burners, sprayers, have a working pressure of 400 lb. per sq. in., the safety valves lifting at 405 lb. per sq. in. Melesco superheaters give a maximum steam temperature of 700 deg. F. Using special Navy boiler compound a boiler clean need be undertaken only every six months intead of the usual two months.

The arrangement of the machinery spaces is interesting. They give an impression of simplicity and of space having been used to the best possible advantage; the novel (to the Royal Navy) practice of mounting the boilers and turbines side by side is convenient and tidy. Despite the operational convenience, there is no reduction in numbers or rank of personnel, just as many ratings being on watch together as there would have been with separate boiler and engine rooms.

Machinery Space Layout.

Taking the forward machinery space, which is the controlling engine-room, the two boilers are off the center line of the ship, slightly to port, facing each other, with two oil fuel pumps on the port side of the boiler footplates and the main and auxiliary feed pumps on the starboard side on about the center line of the ship. On the port side, from forward to aft, are two evaporators with distilling pump between, a low-pressure air compressor, and a turbo-generator set. On the starboard side of the space is the main geared turbine set with a 1000-ton salvage pump, a steam-driven extraction pump, a motor-driven extraction pump, and the main circulating pump under the high control platform forward; two forced-lubrication pumps are aft on the same side (one motor and the other steam) and a feed transfer pump near the hull side. There are also two fire and bilge pumps, one on each side of the ship. The after engine room is similar but transposed *en echelon*.

When steaming, these machinery spaces are very hot. The reason for this is that they are boiler rooms grudgingly conceding space to turbines rather than engine rooms also accommodating boilers. Consequently, while the temperature of the air on the boiler footplates is 112 deg. F. normally, rising to 118 deg. F. maximum, the air high up on the turbine control platform has a normal temperature even higher than this. During the first trials of some of the ships of this class contractor's engineers are said to have found conditions trying, and when they made their first trips through the tropics the ventilation left much to be desired. Comparatively simple modifications effected improvement and it is understood that in later ships modification to the ventilation was made and ambient temperatures reduced as a consequence.

Auxiliaries in other machinery spaces and elsewhere in the ship include diesels for driving generators, Hp. air compressors, CO_2 refrigeration machines, air conditioning plant, etc. The only reciprocating pumps seem to be the oil fuel transfer pumps in the starboard auxiliary machinery space. The impulse turbines driving 250 Kw. auxiliary dynamos turn at 6500 Rpm. and the Allen diesel dynamos have an output of 200 Kw. Each evaporator has an output of five tons of water per hour when the coils are clean. Some 100 tons of water a day is used for domestic purposes and about 20 tons a day when at sea for feed water. There is a diesel engine for providing power for auxiliary steering. Some of the fans are steam driven and some electric, again so as to have alternative sources of supply. The galleys and bakeries are electric.

"Colossus" and Other Classes.

The twenty small fleet aircraft carriers under discussion are properly divided into five classes, ranging from light to intermediate types. The first seven ships, namely the *Colossus, Glory, Ocean, Theseus, Triumph, Venerable* and *Vengeance,* constitute the *Colossus* class. They were laid down by Vickers-Armstrongs (on the Tyne), Harland & Wolff, Alexander Stephen, Fairfield, Hawthorn Leslie, Cammell Laird, and Swan Hunter respectively, between June and December, 1942; they were launched during September, 1943, to October, 1944, and completed from December, 1944, to 1946. With a standard displacement of 14,000 tons (17,720 tons full load) they have a length of 694½ ft., a beam of 80⅓ ft. and a draught of 23 ft. They have a capacity of 39 to 44 aircraft and are armed with numerous pompoms and 20 mm. anti-aircraft guns. They have no armor protection. The starboard island rising from the flight deck, comprising a small funnel, light tripod mast and narrow bridge superstructure, rises from an overhanging sponson, thus allowing practically the whole width of the flight deck to be used throughout its length.

Two somewhat similar ships, the *Magnificent* and *Warrior,* both built by Harland & Wolff Ltd., Belfast, and launched on November 16 and May 20, 1944, respectively, have been transferred to the Royal Canadian Navy. With a displacement of 14,000 tons, they have a length of 650 ft. at the waterline and 690¼ ft. overall, their beam being 80 ft. (waterline) and 112½ ft. (overall). They carry 40 aircraft and are armed with pompoms and 40 mm. Bofors A.A. guns. They have a complement of 1000 officers and men.

The maintenance aircraft carriers *Perseus* (ex *Edgar*) and *Pioneer* (ex *Mars*), both built by Vickers-Armstrongs (the former on the Tyne and the latter at Barrow), were intended to be additional units of the *Colossus* class when laid down on June 1 and December 2, 1942, respectively, but were launched on March 26 and May 20, 1944, and completed in August and February, 1945, as "aircraft maintenance ships" and are apparently modified versions of the *Unicorn.* They have precisely the same displacement, dimensions, armament and propelling machinery as the seven units of the *Colossus* class and a similar appearance with the exception of the prominent square superstructure erected on the flight deck aft.

Later Ships.

Five light fleet aircraft carriers laid down in 1942-43 and launched between September, 1944, and September, 1945, the *Hercules* (Vickers-Armstrongs on the Tyne), *Leviathan* (Swan Hunter), *Majestic* (Vickers-Armstrongs, Barrow), *Powerful* (Harland & Wolff), and *Terrible* (H.M. Dockyard, Devonport), constitute the *Majestic* class. The differences between these ships and those of the *Colossus* class are not perceptible as regards external appearance, but as none is yet quite complete there may be some later. Modifications are largely confined to internal accommodation on the mess decks. All are insulated for tropical service and air-conditioned, the ships' sides being sprayed with asbestos cement instead of being lagged. The watertight doors are of a smaller, improved pattern, being of the Mechan pressed type, with steam heating round the weather doors to prevent the clips icing up in the Arctic. Leading from the cable deck to the forefoot is a downcomer pipe similar to a hawsehole for securing the paravane shoe taking the paravane back hawl. Accelerator gear, similar to the catapult formerly fitted in cruisers, will be installed forward on the port side of the flight deck, which will be reinforced to take larger aircraft than the machines for which this class was originally designed. As far as possible all power capacities are split to allow of alternative supply in the case of failure of one source, some of the

auxiliaries being steam and others electrically driven. The complement of this class has been provisionally estimated at 1343 officers and men. The *Leviathan* was towed from Newcastle to H.M. Dockyard, Portsmouth, in July. Structurally she is almost 100 per cent complete, but completion of fitting out will take some time. The *Hercules* is not so far advanced and she has been laid up at Dalmuir.

"Hermes" Class.

Laid down in 1944, the *Albion* (Swan Hunter), *Bulwark, Centaur* (both Harland & Wolff) and *Hermes* (ex *Elephant*, Vickers-Armstrongs, Barrow), constitute the *Hermes* class. These ships are improved and enlarged versions of the *Majestics*, with a standard displacement of 18,300 tons, and are known as intermediate light fleet carriers. Of those originally ordered, the *Arrogant, Monmouth* and *Polyphemus*, together with the original *Hermes*, have been cancelled.

The maintenance aircraft carrier *Unicorn*, built by Harland & Wolff Ltd., also acted as a light fleet aircraft carrier during the war. Although laid down as a Fleet Air Arm supply and repair ship on June 29, 1939, she was not launched until November 20, 1941, and when completed on March 12, 1943, was equipped to carry 35 aircraft. She was present at Salerno. With a standard displacement of 14,750 tons (20,300 tons full load), she has four Admiralty three-drum boilers and Parsons type geared turbines of 40,000 Shp. on two shafts. She is armed with eight dual-purpose guns, two multiple pompoms and numerous 20 mm. A.A. weapons. She is now equipped for duty as an aircraft repair ship, but in appearance resembles the normal light fleet aircraft carriers for which she probably provided the pattern, although there is one important difference between the two types. The *Unicorn* has a draught of only 19 ft. compared with the 23 ft. of the *Colossus* class, and she has a greater beam to length ratio than the light fleet aircraft carriers of similar displacement. She has ample runway and space for testing aircraft after major repairs.

NEW TYPE MARINE REDUCTION GEAR.

In this article by W. A. Witham of the Western Gear Works, reprinted from "Motorship" for April, 1947, a marine application of planetary gears is described.

A spiral bevel planetary reduction gear for marine use which saves up to seven eighths the space used by the conventional type reduction gear and two thirds the weight has been introduced by Western Gear Works of Lynwood, California, after several years of research and development. Yet all this has been accomplished with no sacrifice in durability or reliability—on the contrary, the new gear has been shown in tests to have longer life and to be able to stand greater strain than a conventional gear unit.

The small size and light weight permits the gear to be placed at a point 45 feet aft of the diesel propulsion engine, just ahead of the stern tube stuffing box, in what is considered to be waste space in the shaft alley.

The location of the diesel engine is the same as with the conventional gear but instead of having the gear directly aft of the engine in the engine room, a high speed, low torque intermediate shaft runs back to the new type gear. This not only reduces the required space but saves considerable dead weight in the shaft itself.

How the gear can be located is shown in the accompanying sketch and is further shown in the actual view aboard the tuna clipper *Sherry Ann*. The saving of space for the gear means more cargo space, a better arranged engine room and better weight distribution.

Ready accessibility for inspection and service is possible as it is only necessary to remove the boards of the catwalk. The gear has its own built-in lube oil pump and reservoir and the only external accessories are an oil pressure gauge and a heat exchanger. A low pressure alarm can be provided which can be installed on the engine control board. The lube oil pump provides pressure lubrication to all the bearings and spray lubrication to the gear teeth.

The planetary gear principle is not new as aircraft engineers have recognized for years the inherently greater load capacity of this type of gear and are using it with all large radial engines for driving the propeller at reduced speeds. For this service, minimum weight is a *must*.

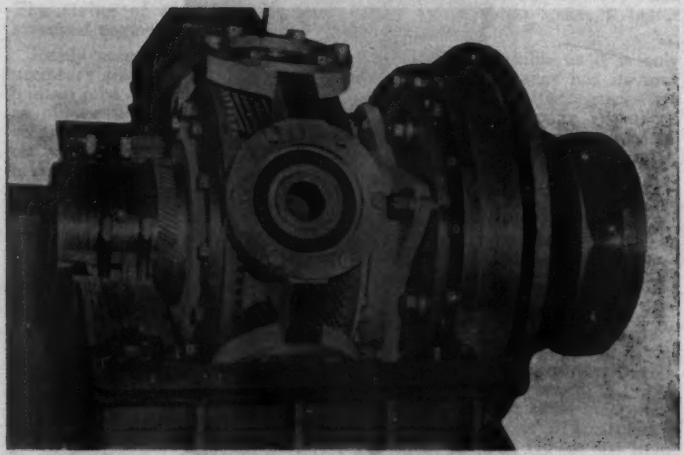
The Western bevel planetary gear uses spiral bevel gears to provide the utmost in smoothness and quietness.

The most striking way to illustrate the potentialities of the planetary gear and the reason for its inherent characteristics of light weight and compactness is by means of a specific application. Thus, consider a conventional parallel axis reduction gear, having a reduction ratio of 2:1 with a pinion pitch diameter of 22 inches, gear pitch diameter of 44 inches and face widths of 18 inches. Using conventional methods of calculating capacity, this unit would be rated at approximately 1000 Hp. at 500 Rpm. Tooth loading would thus amount to 637 lbs. per inch of tooth face.

Now consider a bevel planetary unit for the same power and speed reduction ratio. Assume a pitch diameter of 22 inches for the driving gear, this being the same pitch diameter as the conventional pinion previously mentioned. Using six planet pinions, each having a face width of 3 inches, we have 18 inches of effective face, the same as specified in the conventional gear. Tooth loading is, therefore 637 pounds per inch of tooth face, exactly the same as in the conventional gear. But whereas the conventional gear must be over 44 inches high, over 66 inches wide and substantially over 18 inches long, the bevel planetary unit need be only slightly over 22 inches in diameter and only 18 inches long.

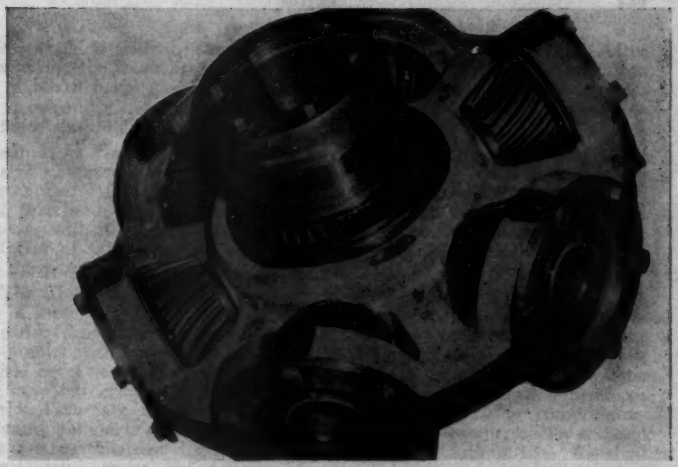
It is customary in conventional offset shaft, helical and herring-bone reduction gear units to heat treat the pinions to a hardness of 210 Brinell, and gears to 180-190 Brinell. This appears to be a practical maximum for gear tooth hardness in gears of this size for the obvious reason that all finish machining must be accomplished after heat treating and the material kept in the freely machinable range in order to maintain the required degree of accuracy. Practically perfect matching of the helices of the gear and pinion must be produced and maintained if the expected load carrying capacity is to be realized. This requires extreme care and precision machining of all control surfaces, such as the journals and mounting surfaces. It also requires large precision hobbing or shaping equipment, and the finest hobs and auxiliary equipment. To maintain the original alignment in service requires a heavy, rigid gear case.

The relatively small diameter and particularly the short faces of the bevel gears used in the bevel planetary unit make it entirely practical to fully harden these gears to utilize the full potential value of the alloy steels regularly used in propulsion gearing. All of the bevel gears used are cut in standard spiral bevel gear generators of moderate size, the steel being preheat-treated at this stage only to provide the best possible machinability. The gears are subsequently carburized and oil hardened, the oil quenching being accomplished in standard type gear quenching



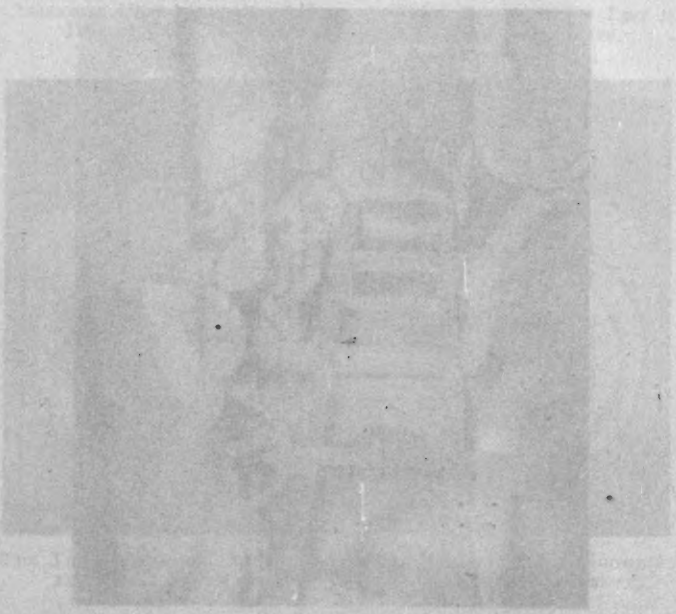
Assembled Unit with Upper Half of Casing Removed. On Left is Driving Gear, Planet Pinions and Mounting in Center, and Stationary Sun Gear on Right.

The Three Principal Components of the Reduction Gear Showing the Short Tooth Faces Which Simplify Manufacture and Also Alignment.



THE PLANET PINIONS AND THE SPIDER. NOTE THE STURDY STRADDLE MOUNTING AND THE USE OF ANTI-FRICTION BEARINGS.

AN INSTALLATION ABOARD THE "SHERRY ANN". THE SMALL SIZE AND IN-LINE FEATURES OF THE GEAR PERMIT USE WHERE CONVENTIONAL GEARS WOULD NOT FIT.



pressures which control, and in fact, practically eliminate distortion. After hardening, the tooth profiles are smoothed up in a short lapping operation in standard bevel gear lapping machines. By this means, tooth surfaces of Rockwell C 60 hardness are produced which are unsurpassed for surface durability and strength.

American Gear Manufacturers Association data in general use in the design of gearing, shows that case hardened gearing has a load carrying capacity of 3½ times the capacity of gears heat treated to 180-210 Brinell. On this basis, the planetary unit with a 22 inch driving gear and six pinions will, if rated at the same capacity as the conventional gear, have a factor of safety at least 3½ times that of the conventional gear, insofar as the gears are concerned.

The life of spiral bevel gears is calculated on the basis of fatigue life by a method developed by the General Motors Research Laboratories and the Gleason Works. By using their formulas and curves, it is possible to predict with exceptional accuracy the fatigue life of the case hardened spiral bevel gears. Unusually high fatigue life is an inherent characteristic of case hardened gears, inasmuch as the hardening operation produces residual surface stresses of the order of 90,000 Psi. in compression. Because fatigue cracks do not start or progress in a compressively stressed surface, case hardened gears have inherently high fatigue life. Residual compressive stress and fatigue life can be increased further by shot peening.

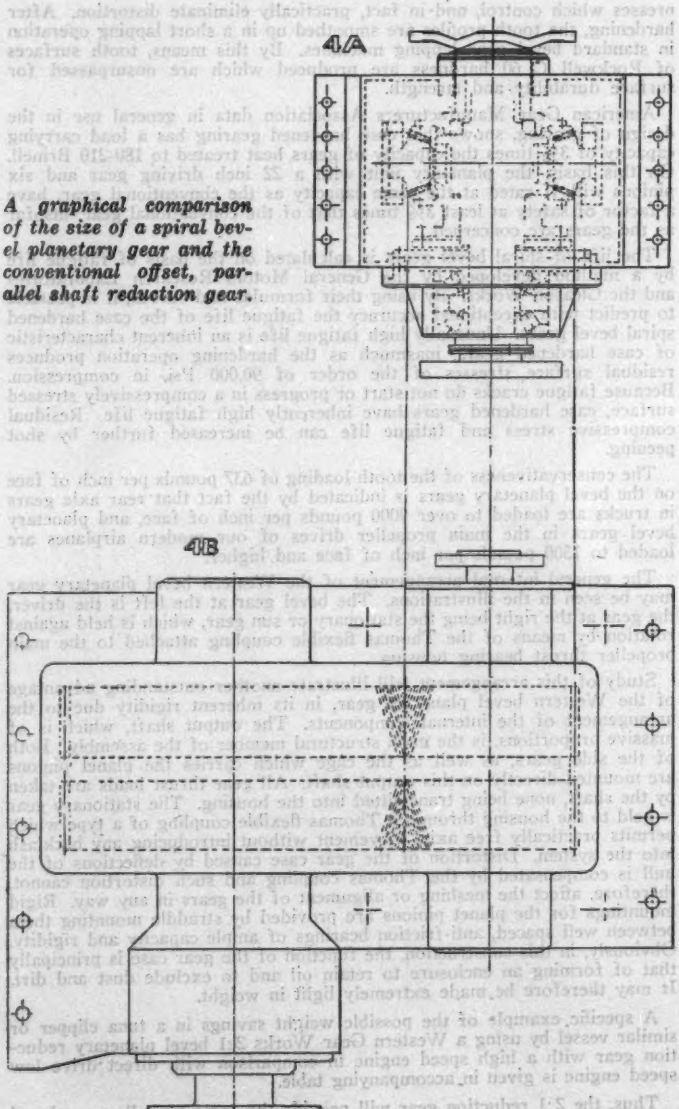
The conservativeness of the tooth loading of 637 pounds per inch of face on the bevel planetary gears is indicated by the fact that rear axle gears in trucks are loaded to over 9000 pounds per inch of face, and planetary bevel gears in the main propeller drives of our modern airplanes are loaded to 1500 pounds per inch of face and higher.

The general internal arrangement of the Western bevel planetary gear may be seen in the illustrations. The bevel gear at the left is the driver, the gear at the right being the stationary or sun gear, which is held against rotation by means of the Thomas flexible coupling attached to the main propeller thrust bearing housing.

Study of this arrangement will illustrate another outstanding advantage of the Western bevel planetary gear, in its inherent rigidity due to the arrangement of the internal components. The output shaft, which is of massive proportions, is the main structural member of the assembly. Both of the side gears, as well as the cage which carries the planet pinions are mounted directly on this output shaft. All gear thrust loads are taken by the shaft, none being transmitted into the housing. The stationary gear is held to the housing through a Thomas flexible coupling of a type which permits practically free axial movement without introducing any backlash into the system. Distortion of the gear case caused by deflections of the hull is compensated by the Thomas coupling and such distortion cannot, therefore, affect the meshing or alignment of the gears in any way. Rigid mountings for the planet pinions are provided by straddle mounting them between well spaced, anti-friction bearings of ample capacity and rigidity. Obviously, in this construction, the function of the gear case is principally that of forming an enclosure to retain oil and to exclude dust and dirt. It may therefore be made extremely light in weight.

A specific example of the possible weight savings in a tuna clipper or similar vessel by using a Western Gear Works 2:1 bevel planetary reduction gear with a high speed engine in comparison with direct drive low speed engine is given in accompanying table.

Thus, the 2:1 reduction gear will provide the same propeller speed and the same torque at the propeller with a saving of 30,500 lbs.



A graphical comparison of the size of a spiral bevel planetary gear and the conventional offset, parallel shaft reduction gear.

Weight Comparison.

Direct Drive Engine.	
6 cylinders, 16 in. by 20 in.	
Rating: 1200 Hp. at 300 Rpm	80,000 lbs.
Reduction Gear Engine.	
8 cylinders, 12 in. by 20 in.	
Rating: 1200 Hp. at 600 Rpm	45,000 lbs.
Saving in Engine Weight	35,000 lbs.
Weight of Western Bevel Planetary Gear.	
2:1 ratio	4,500 lbs.
New Weight Saving per Engine	30,500 lbs.

DISTILLING-PLANT ECONOMY

In this article, reprinted form "Mechanical Engineering" for May, 1947, Mr. Impagliazzo of Griscom-Russell Co., discusses the relative economy of various types of evaporator plants currently in use.

Recently published papers have renewed interest in distilling-plant economy, and in comparisons of the relative economy of the compression-type plants with the conventional steam-heated types.

Compression Distillation.

The principle of compression distillation dates back to the nineteenth century. Periodically, advances in the art of its applications have given it a temporary popularity which then has disappeared as a result of the development of more economical arrangements of conventional types. These advances have generally been stimulated by extraordinary wartime conditions. A plant using a steam-jet compressor was built during the first world war and was tested at the U. S. Naval Engineering Experiment Station at Annapolis. A report on the results of the tests was presented by M. C. Stuart in 1918. "Performance" approximating 3-effect operation of conventional-type plants was reported. The most recent development in this field employs a mechanical compressor, and its arrangement is shown in Fig. 1. "Performance" ranging

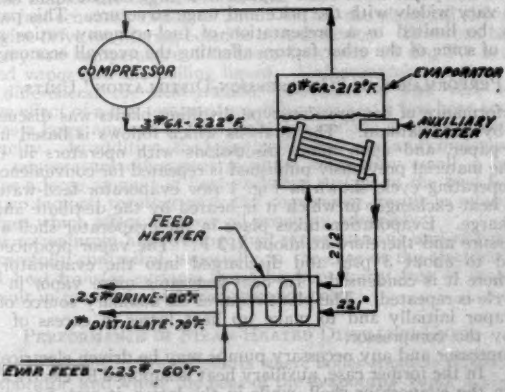


FIG. 1 TYPICAL COMPRESSION-TYPE DISTILLING PLANT

from 5 to 15-effect operation, depending upon the type of drive, is claimed for this unit. This high "performance" has led to some erroneous conclusions regarding the economy of these units.

In order to avoid the confusion regarding terms which exists in the distilling-plant literature, the following definitions will be adhered to strictly in this paper:

"Steam-economy ratio" or simply "steam ratio" means pounds of distillate per pound of heating steam.

"Fuel-economy ratio" or simply "fuel ratio" means pounds of distillate per pound of fuel, based upon fuel having a heating value of 18,500 Btu. per lb.

"Over-all economy ratio" means total cost (including cost of fuel, interest, amortization, care, and maintenance) of producing a unit quantity of distillate.

The steam ratio, very inappropriately called "efficiency" in distilling-plant literature, is useful only in comparing the performance of two steam-heated plants using the same, or a similar, source of steam. If the steam supply differs, two plants having the same steam ratio may have widely different fuel and over-all economy ratios. For example, consider two single-effect distilling plants both heated by steam at 5 psig; assume the first to be heated by exhaust from steam-driven auxiliaries while the second employs steam direct from a boiler. Both may have a steam ratio of 0.95, but if the heat of the auxiliary exhaust would otherwise be wasted, no additional fuel is required to operate the first distilling plant, and its fuel ratio is infinite. The fuel ratio for the second plant is approximately 15.

A comparison of the fuel ratio of the compression-type plant with that of the second steam-heated plant in the previous example permits the conclusion that the compression-type plant gives results comparable to a multiple-effect conventional-type plant. By the same reasoning, however, it may be stated that the first steam-heated plant results in a "performance" comparable to an infinite number of effects. There are very few industrial installations where the heating steam is taken directly from the boiler; hence the "performance" of practically all single-effect installations is equivalent to many effects. It is obvious that "performance" comparisons of this nature can become very confusing.

The over-all economy ratio provides the best means for comparing the performance of different types of plants or of similar plants differently arranged. The fuel-economy ratio is one of the factors affecting the over-all economy ratio, but not necessarily the controlling factor. The presentation of quantitative over-all economy ratios for various types of plants is beyond the scope of this paper since these depend to a large extent upon local conditions, and vary widely with the price and wage structure. This paper must necessarily be limited to a presentation of fuel-economy ratios and to a discussion of some of the other factors affecting the over-all economy.

Performance of "Compression-Distillation" Units.

The performance of compression-type distilling plants was discussed fully and ably by Mr. Latham. The analysis which follows is based upon Mr. Latham's paper, and upon many discussions with operators in the field. Some of the material previously published is repeated for convenience.

In the operating cycle shown in Fig. 1 raw evaporator feed-water passes through a heat exchanger in which it is heated by the distillate and by the brine discharge. Evaporation takes place in the evaporator shell at atmospheric pressure and therefore at about 212 F. The vapor produced then is compressed to about 3 psig and discharged into the evaporator heating element where it is condensed. Its heat generates more vapor in the shell and the cycle is repeated. The plant requires an auxiliary source of heat to produce vapor initially and to make up heat losses in excess of the heat supplied by the compressor.

The compressor and any necessary pumps may be driven electrically or by an engine. In the former case, auxiliary heat is supplied by electrical heating elements in the evaporator shell, while in the latter, it may be supplied by waste heat from the engine in a heat exchanger not necessarily within the shell.

The best performance of these units is illustrated by the following heat balance based upon the quantities and temperatures of Fig. 1:

	Heat in, Btu.	Heat out, Btu.
Evaporator feed*	35	
Compressor	35	
Auxiliary heater	35	
Distillate*		38
Brine*		12
Radiation, venting and other losses		55
	105	105

*Heat content above 32 F.

A heat input of only 15 Btu. per lb. of distillate would be required at the compressor for ideal adiabatic compression through the range indicated, but a minimum of about 35 Btu. is required in the actual unit.

Within rather wide limits, compressor efficiency is of no importance in the case of the electrically driven unit inasmuch as any increase in heat supplied to the compressor is accompanied by a reduction in heat required by the auxiliary heater and vice versa. A total of about 70 Btu. per lb. of distillate must be supplied by electrical energy in any case. Assuming an over-all power-plant thermal efficiency of 25 per cent, the fuel must supply 280 Btu. This results in a fuel-economy ratio of 66.

In the engine-driven unit, where waste heat is utilized to supplement that supplied by the compressor, a fuel-economy ratio of about 160 is possible. A reduction in compressor efficiency has a direct effect on this ratio, however, since auxiliary heat comes from a source that would otherwise be wasted, and therefore a reduction in the amount required does not effect any saving to compensate for the increased amount of energy required by the compressor.

Scale on the evaporator-tube surfaces affects the fuel-economy ratios for the two types of drive in the same manner as compressor efficiency. As scale accumulates, the heat-transfer coefficient is reduced. In order to maintain the same output, the temperature difference across the evaporator tubes must be increased. This means that the vapor must be compressed to a higher pressure; hence more energy must be supplied to the compressor. This has little effect on the fuel-economy ratio of the electrically driven unit but does reduce that of the engine-driven one.

Average fuel-economy ratios of about 60 for the electrical drive and 140 for the engine drive may be expected in actual service. These can be obtained only at very low temperature differences (not over 15 deg F.) between the compressed vapor and the boiling liquid. Unfortunately, at these low temperature differences, the heat-transfer coefficient is also very low. The cumulative effect of a low temperature difference and a low heat-transfer rate necessitates a large amount of heat-exchange surface in the evaporator for a given capacity. In addition, the brine outlet-feed inlet and condensate outlet-feed inlet temperature differences must be made very small, a fact which necessitates a large amount of heat-exchange surface in the feed heater.

These two factors, together with the compressor, result in relatively high fixed charges which affect the over-all economy adversely.

In order to maintain high average fuel ratios, venting and radiation losses must be carefully controlled, especially in the case of the electrically driven unit. All heat-transfer surfaces must be kept fairly free of scale. These factors result in relatively high maintenance costs.

Performance of Steam-Heated Distilling Units.

For single-effect steam-heated plants where boiler steam is supplied directly to the evaporator, and where the heat of the evaporator vapor is absorbed by raw circulating water, the fuel ratio is approximately 15. Even multiple-

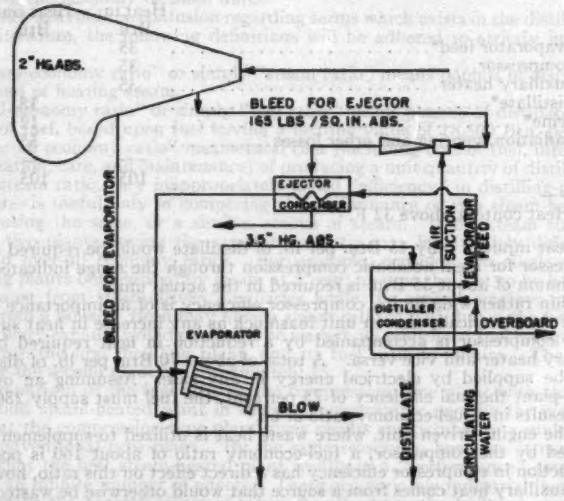


FIG. 2 TYPICAL SINGLE-EFFECT LOW-PRESSURE DISTILLING PLANT

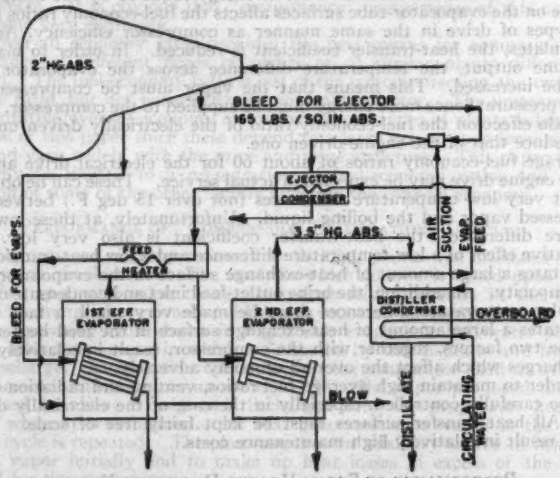


FIG. 3 TYPICAL TWO-EFFECT LOW-PRESSURE DISTILLING PLANT

effect plants operating between the same limits result in relatively poor fuel economy.

Unusually high fuel ratios can be obtained, however, by using low-pressure bleed steam to operate independent vacuum-type distilling plants, as illustrated in Figs. 2 and 3, or by making use of a "reducing-valve" evaporator. The term, "reducing-valve" evaporator, has usually been used to designate only those units which actually replace a reducing valve, i. e., where steam at a given pressure is condensed in the evaporator, and the evaporator vapor at a lower pressure is then utilized in some industrial process. It is proposed here to use the term in a broader sense so as to include any evaporator so tied in with other plant operations that the lower-pressure evaporator vapor is returned to the system to take the place of the steam used by the evaporator. Thus evaporators used in series with bleeder heaters in power plants (illustrated in Fig. 4) will fall in this general classification.

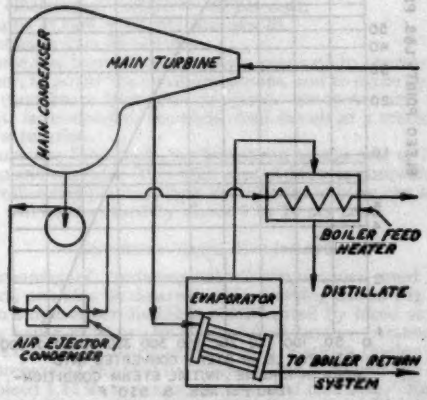


FIG. 4 REDUCING-VALVE TYPE EVAPORATOR FOR STEAM POWER PLANTS

In order to analyze the performance of "reducing-valve" evaporators and low-temperature distilling plants operated by bleed steam, turbine performance must be known. Fig. 5 indicates approximately the amount of heat per pound of steam converted into mechanical energy in expanding from an initial pressure of 1200 psia, and a total emperature of 950 F., to a final pressure of 2 in. Hg abs. A turbine engine efficiency of 80 per cent has been assumed.

A total of 458 Btu. per lb. are converted into mechanical energy in the complete expansion. If the steam is bled from the turbine at 10 psia, 344 Btu. are converted into work, a reduction of 114 Btu. This means that the increase in steam to be produced at the boiler is only (114/458) or 0.249 lb. for every pound bled in order to maintain the same turbine energy output. If the condensate from the bled steam is returned to the boiler-feed circuit with only 10 deg subcooling, it will be at a temperature of 183 F. instead of the 101 F. in the main condenser, and it is fair to reduce the 0.249 lb. slightly to compensate for the higher drain temperature. The equivalent is found to be actually about 0.23 lb. In a similar manner, the equivalent increase at the boiler per pound of steam bled at any point can be determined.

Independent Low-Pressure Plants.

In the low-pressure distilling plants shown diagrammatically in Figs. 2 and 3, steam is bled from the turbine at about 165 psia to operate the air ejector (and any necessary pumps), and from a lower pressure to supply the necessary

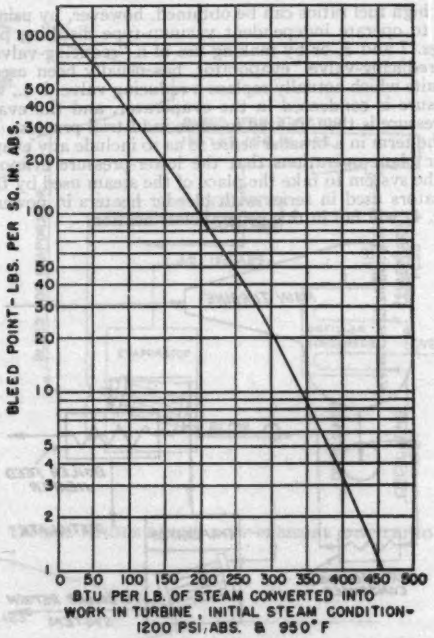


FIG. 5 APPROXIMATE BTU PER POUND STEAM CONVERTED TO WORK

(Expansion from 1200 psia and 950 F.)

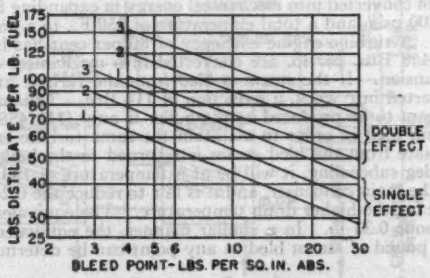


FIG. 6 FUEL ECONOMY OF LOW-PRESSURE BLEED STEAM-HEATED PLANTS

heat for evaporation. From the distilling-plant heat balance, the amount of steam to be bled at each of these points per pound of distillate may be determined. These values may then be converted to their equivalents at the boiler by the application of proper factors. The amount of fuel required to produce this additional steam at the boiler, and to supply also the further additional steam to preheat it in accordance with the appropriate power-plant heat balance, then may be determined.

Fig. 6 shows the results of such calculations for three typical power plants. This chart is based upon the following conditions:

(*a*) Initial steam conditions at turbine throttle:
Curve 1 415 psia, total temperature = 700 F.
Curve 2 615 psia, total temperature = 900 F.
Curve 3 1200 psia, total temperature = 950 F.
(*b*) Turbine exhaust = 2 in. Hg abs.
(*c*) Engine efficiency of turbine = 80 per cent.
(*d*) Heat value of fuel = 18,500 Btu. per lb.
(*e*) Boiler efficiency = 85 per cent.
(*f*) Allowance has been made for steam required to operate the distilling-plant air ejector, to power the necessary pumps, and to make up all heat losses.
(*g*) Steam condensate from the air-ejector condenser and from the first-effect tube nest is returned to the boiler-feed circuit at a temperature reasonably close to saturation.
(*h*) Allowance has been made for blowdown at the rate of 200 per cent of the distilling-plant output. This high rate of blow is desirable in sea-water plants. In fresh-water plants, where the amount of blow can be reduced, the fuel economy will be increased by about 5 to 10 per cent.

Reducing-Valve Evaporators.

The fuel economy of "reducing-valve" evaporators, used in series with bleeder heaters, can also be determined by making use of Fig. 5. Consider, for example, a single-effect distilling plant heated by bleed steam at 20 psia and delivering vapor at 10 psia to a boiler-feed heater. Assuming the evaporator produces 0.90 lb. of distillate per pound of steam (higher values may be obtained by preheating the feed by means of the blow and the distillate), it is necessary to bleed 1.11 lb. of steam per pound of distillate. Each pound bled at 20 psia is equivalent to (153/458) or 0.334 lb. at the turbine inlet, and therefore the flow through the turbine must be increased by 0.334×1.11 or 0.372 lb. for every pound of distillate. However, 1.02 lb. of steam would have to be bled at 10 psia in any event in order to accomplish the same amount of boiler-feed heating obtained per pound of distillate. This would be equivalent to $1.02 \times (114/458)$ or 0.254 lb. of steam at the turbine inlet per pound of distillate. Furthermore, the 1.02 lb. would be returned to the system at about 190 F. instead of about 220 F. actual drain temperature from the evaporator heating element. An additional flow of 0.01 lb. would be required through the turbine to compensate for this difference in drain temperatures. Thus the net increase in steam flow through the turbine per pound of distillate is only $0.371 - (0.254 \times 0.01)$ or 0.107 lb. Depending upon the particular power plant involved, the amount of fuel oil necessary to generate this greater amount of steam will be between 0.0075 and 0.0092 lb.; thus the fuel-economy ratio is between 110 and 132 lb. of distillate per pound of fuel. By a similar process, an evaporator supplied with steam bled at 5 psig and returning vapor at 0 psig to a boiler-feedwater heater will be found to have a fuel ratio between 165 and 190.

The fuel-economy ratio increases as the difference in pressure between the bleed point and the evaporator vapor is reduced. At the same time, however, the amount of evaporator heat-transfer surface required for a given output also increases very rapidly. From the standpoint of distilling-plant economy alone, the optimum pressure difference must be found by balancing the reduction in fuel costs against the resulting increase in fixed costs. In existing power plants of course other factors may also require consideration.

There is little of a general nature that can be said about reducing-valve evaporators providing steam for industrial processes. If the steam is bled at a higher pressure because of the existence of the evaporator, the fuel ratio may be determined in the same manner as for the reducing valve evaporator used in series with a bleeder heater. If, however, the evaporator actually takes the place of a reducing valve in supplying reduced-pressure steam for some industrial process, the fuel ratio depends upon the additional heat required at the boiler to make up the difference between the steam that would have been delivered by the reducing valve and that which is actually produced in the evaporator for the same input. These fuel ratios will generally be higher than those for reducing-valve-type evaporators used in power plants since there is no power loss to be considered.

Relative Performance of Various Types.

In some cases the question of relative over-all economy of the various types of units may be entirely academic because the conditions imposed may make some other criterion more important. Thus, the military may consider mobility more important than economy, or space considerations may dictate a single-effect plant where a double-effect plant might prove more economical, etc. The following general observations, however, are usually applicable:

1. Heat losses from radiation, venting, brine discharge, etc., are of critical importance in the case of compression-type units but only of minor importance in most steam-heated units, and negligible in the case of low-pressure independent plants. Thus an increase in these losses of 20 Btu. per lb. of distillate over those upon which the fuel ratios previously given were based would reduce fuel economy as follows:

	Per cent
Electrically driven compression unit	22
"Reducing-valve" evaporator	7
Low-pressure independent plant	2

2. Because of the negligible effect of heat losses, low-pressure independent plants are especially suitable for use where the quality of the raw water is such as to require a large amount of continuous brine discharge. With sea water, for example, good practice requires a blowdown rate of 200 per cent of the distillate to minimize scale formation. The fuel ratios given for compression-type units were based upon a blowdown rate of only 25 per cent. An increase to 200 per cent would require an enormous feed heat exchanger in order to maintain the same brine-outlet-feed inlet temperature difference, and even then would result in a reduction of about 35 per cent in the fuel ratio. The fuel ratios given in Fig. 6 for low-pressure plants are already based upon a blowdown rate of 200 per cent and no brine-feed heat exchanger whatever is required.

3. Steam-heated plants are generally simpler to operate than vapor-compression-type units. Heat input and heat output automatically balance themselves in steam units whereas it is necessary to achieve this balance by regulating the amount of auxiliary heat supply, or by regulating the amount of brine discharge in the compression-type units. Control of water levels, etc., can be made automatic in both types.

4. In most cases, both the initial and maintenance costs of compression-type units for a given capacity will be greater than those of steam units. A large part of these charges can be attributed to the compressor.

5. Scale formation is practically eliminated by low-pressure low-temperature distillation. When the heating steam temperature does not exceed 160 F. and evaporation takes place at 125 F. or less, operation with untreated feed for a year or more without cleaning is quite possible. At present, such low-temperature distillation is not practical with compression-type units because the size and cost of vapor compressors increase prohibitively as the suction pressure is reduced. Future advances in steam-compressor performance may remove this disadvantage.

6. Good fuel economy can be obtained in the vapor-type compression units with very low temperature differences. At the same temperature difference, the single-effect low-pressure bleed-steam-heated plant yields about the same fuel economy as the engine-driven compression-type unit, while the 2-effect low-pressure plant and reducing-valve evaporators give better results.

7. The cost of engine fuels is considerably higher than that of boiler fuels. This may make the internal-combustion-engine-driven compression-type unit the least economical when the other alternatives are possible.

8. Vapor-compression-type units appear to be economical only in cases where distillation must be accomplished entirely independently of a steam power plant or other steam installation. If there is a cheap source of electricity available, the electrical drive should prove advantageous. The engine-driven compression-type unit is particularly useful where great mobility is desirable.

PROBLEMS IN THE MECHANICAL DESIGN OF GAS TURBINES.

High temperatures involved in the operation of the gas turbine have introduced many new problems in the properties of the metals with which the designer has to work. This paper by Mr. Ronald B. Smith, Vice President of the Elliott Company, is reprinted from the Journal of Applied Mechanics for June 1947 and outlines some of these problems and offers a line of approach taken successfully by the author's company in solving them. (The Elliott Company is under contract with the Navy for the manufacture of an experimental gas turbine).

Machine-Design Concepts Altered With Advent of High Temperatures.

Apart from the problems of manufacture and production, the mechanical design of an engineering structure can generally be reduced to a consideration of two broad technical problems, as follows:

1. An analytical determination of the stresses imposed on the structure as a result of external loads.

2. An evaluation of the stresses in relation to the physical properties of the material of which the structure is composed.

For the greater part, or at least until the advent of the high-temperature turbine, our concept of mechanical design has been closely associated with Hooke's law and yield-point failure. Under these circumstances, the theory of elasticity has provided a means for the computation of stress, and for the vast majority of our materials, the Mises-Hencky condition of maximum strain energy a criterion of failure. From our present point of view, the significance attached to yield-point stress, in evaluating what we now consider a low-temperature design, is the important feature. The structure was designed for elastic yielding only, with the parts so proportioned that the yield-point stress was the ultimate to which the material could be subjected without bringing about distortions that would cause failure.

The inadequacy of Hooke's law in expressing the physical properties of a material at elevated temperature was thoroughly appreciated by designers more than 20 years ago, with the result that an extensive program of creep-testing and analysis was undertaken and, in fact, is still in progress. However, even the realization that stress was a function of time as well as strain did not alter basically our concept of design. Almost without exception we directed our high-temperature creep programs toward the development of materials which would have creep rates less than 10^{-8} in. per in. hr. under the loading imposed; we held to the original premise of designing for minimum possible yielding, and our theory of failure remained essentially that of Mises-Hencky.

These premises have proved entirely satisfactory for the design of machinery operating at temperatures up to 900 F., and while they may be criticized in light of our most recent experiences with gas turbines, they can likewise be

defended on the ground that they tied elevated-temperature machine design to the broad fundamentals which had been established for normal-temperature design.

The advent of the gas turbine for operating service at temperatures in excess of 1000 F. has necessitated a complete change in our design concepts. The materials available for construction yield so continuously to these temperatures that, if they are to be used effectively, all hope for designs in which yielding during the normal life span can be held to 0.1 per cent or less must be abandoned. The designs must be mechanically sound for continuous yielding to the extent of 0.5 to 1 per cent. It is evident that the importance formerly attached to creep is considerably altered, while the Mises-Hencky criterion of yielding under combined stresses is obviously inapplicable. High-temperature machine design, of necessity, has abandoned completely the framework of our previous theory and, in so doing, has pushed the designer out on the same limb he was rescued from some 30 years ago. It is the object of this paper to discuss the present basis of high-temperature machine design as it is practiced by at least one engineering group, with the hope that by emphasizing its short-comings the designer may be more rapidly rescued from the uncomfortable position he now occupies.

Application of Theory of Plasticity.

For the computation of steady stress, the high-temperature machine designer is on a fairly solid ground, as a result of the work already done on the theory of plasticity by Nadai and others. Disks are computed on the basis of an average tangential stress uniformly distributed over the radial cross section, bending stresses are evaluated upon a parabolic rather than a triangular stress distribution, and simple tension is evaluated without regard to stress concentrations. In many respects stress analysis under plastic circumstances is simpler than it is under elastic circumstances; it is in the evaluation of a safe working stress that the greatest uncertainty exists.

Present designs of the author's company are based upon the maximum-stress theory of failure, and the evaluation of working stress is made on the basis of the stress-to-rupture test in simple tension at elevated temperature. A typical set of working data for 19-9DL at 1200 F. is shown in Fig. 1. Not only the stress to rupture but the total elongation versus stress and time is plotted. Working stresses for short-life machines, such as aircraft turbines, are based upon an expected life of 1000 to 1500 hr., whereas power machinery is designed on the basis of 100,000 hr. at full temperature. The stress-rupture tests are generally extrapolated linearly on the semilog plot of Fig. 1, to arrive at probable values of the stress to rupture under these long-time periods.

Fortunately, most of our working data have been extended to include rupture failures for testing times of 10,000 hr., so that the extrapolation generally covers only one logarithmic cycle. The working stress is usually selected for the given period of time, at a value of 50 to 60 per cent of rupture strength, but when serious vibratory stresses are anticipated added safety is introduced. For a working stress of 60 per cent of rupture strength, the total plastic yielding will generally be from ½ to 1 per cent, and the design is made to allow for at least this degree of yielding without disastrous interferences between rotating and stationary members. Where close clearances are required for reasons of efficiency, they are made with thin sealing strips which wear away as yielding occurs.

Maximum-Stress Theory of Rupture.

The maximum-stress theory of rupture is most assuredly open to criticism, but on the basis of present-day evidence, it appears to be the only logical choice. Even for normal temperatures there exists as yet no satisfactory definition of the surface of rupture as a function of the three principal stresses, although the surface of yielding is fairly well prescribed. A critical examination of rupture theory by both engineers and metallurgists is urgently needed.

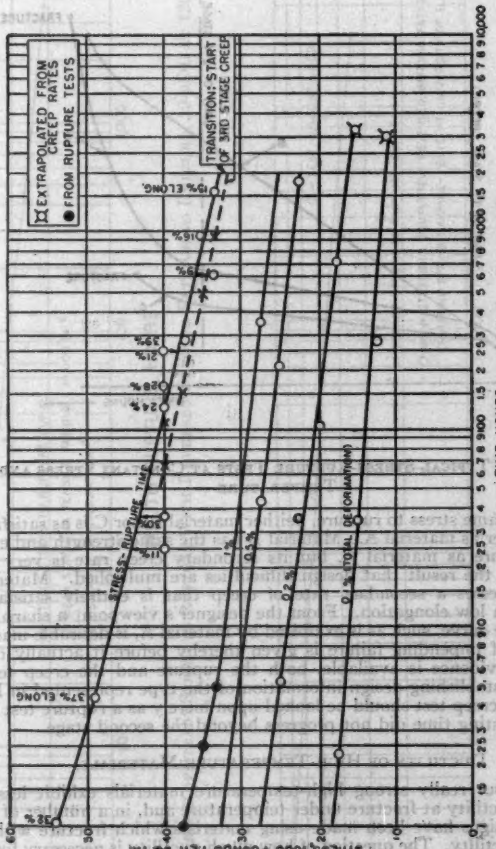


FIG. 1. TYPICAL DESIGN DATA: FORGED 19-9DL AT 1200 F

For high-temperature design, the rupture test should be so conducted that it yields not only fracture time but a continuous record of elongation as well. Typical stress-rupture tests on three materials at constant stress and temperature are illustrated in Fig. 2. While all three of these materials will

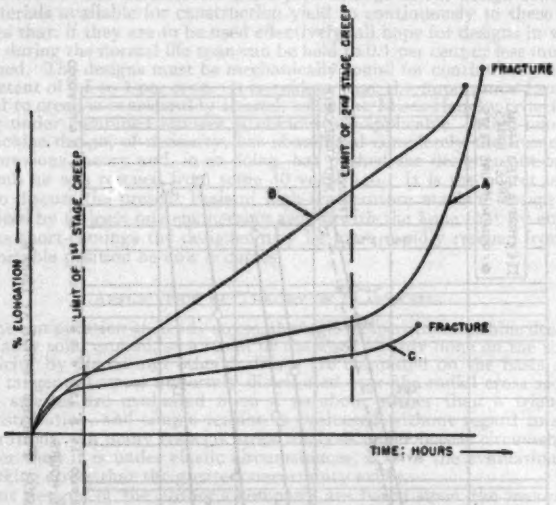


FIG. 2 TYPICAL STRESS-RUPTURE TESTS AT CONSTANT STRESS AND TEMPERATURE

indicate the same stress to rupture, neither material B nor C is as satisfactory to the designer as material A. Material B has the same strength and elongation at fracture as material A, but its secondary creep rate is very much greater, with the result that design difficulties are multiplied. Material C, while it possesses a secondary rate of creep that is entirely satisfactory, fractures with low elongation. From the designer's viewpoint a sharp break in the rupture curve, such as is exhibited by material A, is desirable inasmuch as warning of impending failure is given thereby before it actually occurs. Until more evidence is available, both the rupture and the creep test are valuable in establishing design information of the type represented in Fig. 1, although the creep test should be looked upon merely as a rupture test which within the testing time did not progress beyond the second stage.

DUCTILITY OF HIGH-TEMPERATURE MATERIALS.

Many of our really strong high-temperature materials exhibit less than 5 per cent ductility at fracture under temperature and, in a number of cases, successful designs have been made using materials which fracture with only 1 per cent ductility. The question of how much ductility is necessary for good design obviously needs a re-examination. For fairly uniformly stressed members such as disks, computations show that less than ¼ per cent elongation is ample to redistribute the stresses from an elastic to a plastic state. For members in which concentrated stresses may exist, such as is found in the threads of bolts, or in the roots of broached turbine blading, from 2 to 3 per cent elongation is a necessity if these highly localized stresses are to be redistributed without incipient cracking. Materials which fracture with from 3 to 5 per cent elongation, and with definite evidence of third-stage yielding

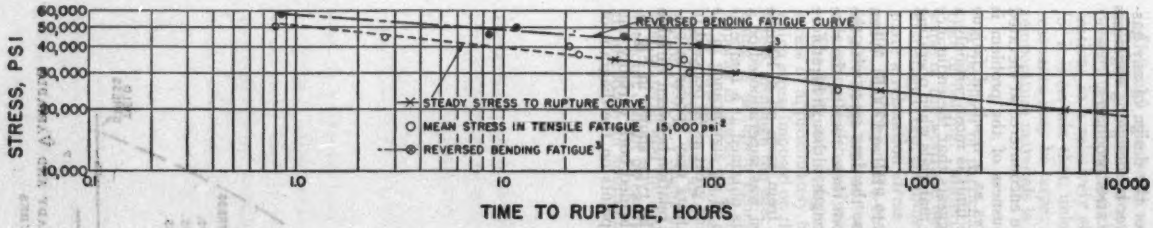


FIG. 3 COMPARISON OF STEADY STRESS TO RUPTURE WITH REVERSED BENDING FATIGUE TO RUPTURE; 19-9DL AT 1200 F

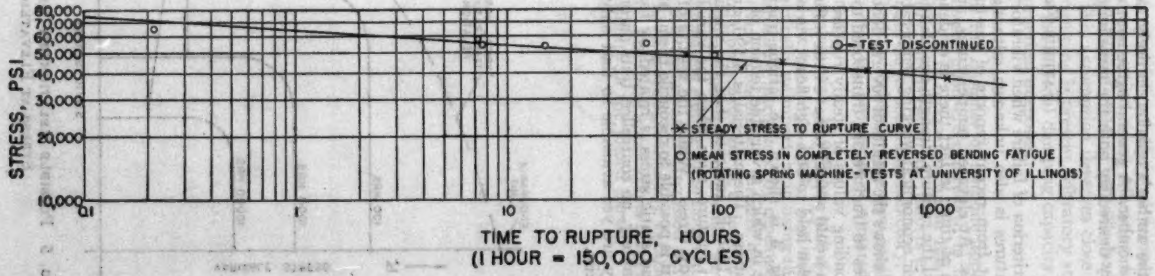


FIG. 4 COMPARISON OF STEADY STRESS TO RUPTURE WITH TENSILE FATIGUE TO RUPTURE AND REVERSED BENDING FATIGUE TO RUPTURE; S-590 AT 1350 F

(Solution-treated bar stock, aged for 16 hr at 1400 F.)

are, in the author's opinion, entirely satisfactory for the design of any gas-turbine member. A general agreement on this matter by designing engineers is greatly desired by both the metallurgists and the steel producers.

Fatigue Fractures.

The criterion of failure when vibratory stresses are encountered at elevated temperatures is also unknown, although the seriousness of the problem is apparent from recent reports of gas-turbine failures as a result of fatigue fractures. At elevated temperatures, an endurance limit is most certainly a function of time; in fact there is evidence which suggests that the endurance limit and the rupture strength are identical for some materials when operating near their economical working temperature.

Fig. 3 shows the results of reverse bending tests on 19-9DL at 1200 F. When the cycles to fracture are transposed to a time basis, the data agree with the corresponding values of steady rupture stress. From these results, the rate of strain would appear to have little influence on strength, although heretofore it has been held to be a significant variable.

In Fig. 4 is shown the fatigue-fracture results from tests on a Krouse machine in which a variable tension of ± 15,000 psi was superimposed on a steady stress. The material is S-590, and the results obtained at a temperature of 1350 F. indicate that the variable stress is of no significance in its effect on rupture, even when the variable stress is as great as 60 per cent of the steady stress. While the results of these two tests may appear incompatible, it is possible to reconcile them, providing the relation between variable stress and static stress is related by functions of the type shown in Fig. 5. Unfortunately, no controlled tests on one material loaded both variably and statically are yet available so that only boundary points on the diagram of Fig. 5 exist.

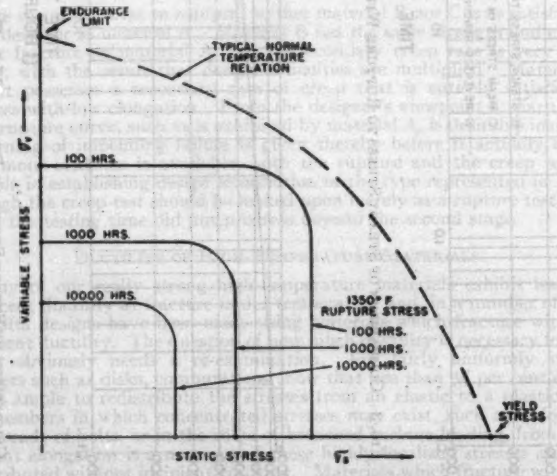


Fig. 5 Possible Relationship Between Steady and Variable Stress at Elevated Temperatures

Although the notch sensitivity of materials at elevated temperature is unknown, the influence of stress concentration when vibrating loads are present is neglected on the basis that localized yielding will alleviate concentrations in a relatively short time. This assumption deserves close scrutiny for it is certain that, unless external damping is present, vibratory stresses at resonance may be large; inasmuch as the internal damping capacity of high-temperature materials is zero for all practical purposes.

Bolting Materials.

A problem of particular concern to the designer of long-life apparatus is that of bolting. Materials which possess good creep strength in the usual sense of the word are not necessarily good in relaxation, for in this case low first-stage creep, combined with high ductility is essential. Efforts to approximate a relaxation stress from creep data have been unsatisfactory, with the result that step-down tests, as described by Robinson, are now considered as important as rupture tests in determining the properties of a material. The relaxation is also analyzed for creep rates.

The bolts are designed for retightening each 10,000 hr. with a working stress of approximately 60 per cent of the relaxation limit. The initial tightening stress imposed on the bolt is practically unimportant in fixing the relaxed stress, although the effect of strain-hardening introduced by repeated retightening is definitely beneficial. Relatively loose-fitting threads have proved superior in respect to galling, to ground closely fitted parts, at least, in 19-9DL type material. Colloidal silver compound has been uniformly satisfactory with temperatures up to 1400 F. in eliminating galling when both nut and bolt are of 19-9DL. Slight differences in the coefficient of expansion between the bolt material and the flange material have proved particularly vexatious when the joint cannot be easily retightened. For these applications it is customary to introduce an additional elastic flexibility in the joined parts.

ASSOCIATION NOTES.

Addresses.

It is urged that each member keep the Society informed as to his correct address.

Society Lapel Button.

Shown below is an illustration of the lapel button of the Society. It is believed that it will be conceded that this is a

very fine, dignified insignia. It is one-half inch in diameter. The oak leaves and lettering are red on a gold back-ground. It is available to all members at fifty cents (50c) each.

Annual Meeting and Nominations to Offices for 1948-1949.

In accordance with its By-Laws, the Annual Meeting of the Society was held in Washington, D. C., on Tuesday 7 October, 1947. The following nominations for President and Secretary-Treasurer for 1948, and for Council Members for 1948-1949 were made:

For President:

Rear Admiral R. W. Paine, U.S.N.

For Secretary-Treasurer:

Captain F. W. Walton, U.S.N.
Captain J. E. Hamilton, U.S.N.

For Member of Council:

Rear Admiral J. W. Fowler, U.S.N.
Rear Admiral T. C. Lonnquest, U.S.N.
Captain L. V. Honsinger, U.S.N.
Captain Rex L. Hicks, U.S.N.
Captain W. N. Mansfield, U.S.N.R.
Lieut. Commander Nicholas Tiedeman, U.S.N.R.
Mr. E. C. Magdeburger
Mr. M. Douglas Gibson, Jr.

Ballots have been distributed. Polls close at 3:30 P.M. on Friday, 26 December, 1947.

Amendment To By-Laws.

The following amendment to the By-Laws has been proposed:

Paragraph 9—Amend by the addition to the sentence "In the temporary absence of the President, the senior member present of the Council of the three branches of the Service shall act as President"

so that the paragraph shall read:

Paragraph 9—"The President shall excercise the usual duties of that office. In the temporary absence of the President, the senior member present of the Council of the three branches of the Service shall act as President"

Provision for voting on this proposed amendment has been made on the ballot for offices.

Prize Essay For 1947-1948.

Technical essays are invited from any member of the American Society of Naval Engineers, any Navy or Coast Guard officer or enlisted man, regular or reserve, in competition for the 1947 essay prizes. In order not to be restrictive in subject matter, the Council will consider all papers which fall within the scope of paragraph 2 of the by-laws, the pertinent part of which is here quoted:

"The object of the Society shall be to promote a knowledge of marine engineering and Naval architecture by reading, discussing and publishing papers on professional subjects; by bringing together the results of experience acquired by engineers in all

parts of the world and publishing them in the JOURNAL of the Society; by publishing the results of such experimental and other inquiries as may be deemed of value to the advancement of the science; and by recording historical events in the lives of engineers."

The contestant may select any subject consistent with the requirements of the foregoing papargraph. The Council considers subjects directly affecting marine engineering and Naval architecture, e.g., corrosion resistance, electronics navigational aids, metallurgy, etc., to be within the scope of the by-laws provided the application to ships is considered in the essay. Considerable latitude in treatment of a subject is permissible. For example, if a contestant is particularly conversant with one phase of a problem, a paper confined to that one phase would not be ruled out. Supporting engineering data may or may not be included at the option of the contestant.

Articles must be received by the Secretary-Treasurer, American Society of Naval Engineers, Navy Department, Washington, D. C., by March 1948. One prize of $500, one of $300 and one of $200 may be awarded for papers considered by the Council to be worthy of such awards. Those papers awarded prizes will be considered the property of the Society and may be published in the JOURNAL without further payment. If so published, due credit will be given to the author. If other papers submitted in the contest are published in the JOURNAL, they will be paid for at the prevailing publication rates. All other papers will be returned.

In judging articles, first consideration will be given to technical contents. English composition, originality of contents, analysis of problems presented, and possibilities of practical application of conclusions reached are also factors to be considered.

The length of the article is not restricted, but one in the order of 5000 words is suggested.

Any article submitted in accordance with the above rules must be plainly marked to indicate that it is submitted in competition for the Prize Essay Award.

SCHOLARSHIP.

The Council announces the continuation of the Scholarship at Webb Institute of Naval Architecture, Crescent Beach Road, Glen Cove, Long Island, N. Y.

The Scholarship provides for total expenses for candidates who qualify for entrance.

Eligibility for qualifying for Scholarship and entrance are:

a. Be a full dependent of Member of the Society (Male).

b. Be between 16 and 21 years of age at time of qualifying examinations (June of year of entrance).

c. Be an American citizen.

d. Be unmarried.

e. Be acceptable to the Institute and Society (demonstrated by competitive written and oral examinations).

f. Applicants must have at least 16 credits from an accredited high school or equivlaent, including the following:

4 years of English
2 years of a foreign language
1 year of Physics
2 years of History (1 year of which must be American History)
1½ years of Algebra
1 year of Plane Geometry
½ year of Solid Geometry
½ year of Plane Trigonometry

Successful completion of the course at the Institute leads to a degree of Bachelor of Science.

The Institute will arrange for conducting examinations in the locality of the applicant's residence. Examinations will be conducted during June and entrance follows in August (specific dates will be included in individual notices).

Applicants must make application to Secretary-Treasurer, American Society of Naval Engineers direct, not later than 1 May 1948.

MEMBERSHIP.

Attention of all non-member officers is especially invited at this time to the advantage and benefit to be attained by membership in this Society. The Society was organized in 1888—for the sole purpose of the advancement of Engineering. Its JOURNAL, published quarterly, in February, May, August and November, and completing its 59th year of continuous publica-

tion, has attained a foremost rank in its field and is a recognized authority throughout the engineering world. It is used generally by Naval officers and civilians alike as reliable reference book. The Bureau of Ships considers the JOURNAL of such value that it makes it available for the libraries of all Naval vessels and shore activities of any size. The presence of the JOURNAL in your personal library cannot fail to be of great benefit to you, often making available information of great value which can be obtained from no other source.

It is particularly suggested that officers of the Naval Reserve, who are returning to civil life, should arrange for membership prior to detachment from active duty.

Annual dues are $5.00, which are probably lower than those of any other technical Society. There is no initiation fee, nor any extra charge for the JOURNAL. This extremely low cost is possible only because the Society is bona fide operated solely for your benefit without any pecuniary profit to any one.

An application blank generally will be found at the back of each copy of the JOURNAL. However, if none is found, application by letter will be accepted.

It is suggested that Members will confer a real favor on non-members by bringing the Society to their attention.

The Annual Meeting of the

Society for Experimental Stress Analysis

will he beld at

The Hotel Pennsylvania, New York, N. Y.

on

December 4, 5, 6, 1947

Inquiries should be addressed to the Society for Experimental Stress Analysis, P.O. Box 168, Cambridge 39, Massachusetts.

MEMBERSHIP ADDITIONS.

The following have joined the Society since the publication of the August, 1947, JOURNAL. Addresses of personnel on ships not given:

NAVAL.

Brown, John B., Captain, U.S.N.,
Bureau of Ships, Navy Dept., Washington, D. C.

Carson, Alfred Joseph, Lieut., U.S.N.R.,
214 E. Gilpin Ave., Norfolk, Va.

Duran, August J., Lieut. Commander, U.S.N.R.,
Mech. & Elec. Power Engineer, & Marine Engineer
175 N. Victoria St., St. Paul Minn.

Foster, Richard E., Lieut. Commander, U.S.N.,
Code 665, Bureau of Ships, Navy Dept., Washington, D. C.

Kasl, James M., Lieut., j.g., U.S.N.R.
Engineering Dept. The Ocean Accident and Guarantee
Corp., Ltd. 1727 Federal Reserve Bank Building,
Kansas City, Mo.

Gaasterland, C. L., Commander, U.S.N.
U. S. Submarine Base, Navy No. 128, c/o Fleet P.O.
San Franscisco, Calif.

Griffin, John J., Lieut., U.S.N.R.,
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Kell, Walter M., Commander, U.S.N.R.,
442 South George St., York, Pa.

Mann, Everett E., Captain, U.S.N.,
Naval Training Schools, Mass. Inst. of Technology
Cambridge 39, Mass.

Millar, Donald, B., Lieut., U.S.N.

Muller, Herbert S., Lieut., j.g., U.S.N.,
254 Roff Ave., Palisades Park, N. J.

Sawyer, John W., Lieut. Commander, U.S.N.R.,
Code 641, Bureau of Ships, Navy Dept., Washington, D. C.

Sheldon, Frank M., Lieut., U.S.N.

Terhune, John A., Commander, U.S.N., Ret.
1000 Second St., Coronado, Calif.

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Mail 1190 East 152d St., Cleveland, Ohio.

Matiuk, Alexander, Chief Mech. Engineer, Douglas M. McBean, Inc., Construction Engineers, Rochester, N. Y.
Mail 597 East 53d St., Brooklyn, N. Y.

ASSOCIATE.

Barnicle, J. Lester, Traffic Representative, Mississippi Shipping Co., 255 Transportation Building, Washington, D. C.

Juhl, Vernon S., Planner & Estimator, Planning Branch, Planning Division, Pearl Harbor Naval Shipyard, Pearl Harbor, T.H.

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